G 1087 EPRI-95-02

EPRI

NEXT GENERATION GEOTHERMAL POWER PLANTS

EPRI Project WO3657-01
Final Report
September 1995

Prepared by

CE Holt Company
201 South Lake Avenue
Pasadena, California 91101

Prepared for

Electric Power Research Institute
3412 Hillview Ave
Palo Alto, Ca 94304

Renewables and Hydro Business Unit
Generation Group

EPRI Project Manager
Dr. Evan Hughes

DISCLAIMER

NOTICE

This report has not been reviewed to determine whether it contains patentable subject matter, nor has the accuracy of its information or conclusions been evaluated. Accordingly, the report is not to be considered a published report and is not available for general distribution, and its distribution is limited to employees and advisors of EPRI for the sole purpose of evaluating the progress and future course of the project described in the report. Until the report has been reviewed and evaluated by EPRI, it should be neither disclosed to others nor reproduced, wholly or partially, without the written consent of EPRI.

Next Generation Geothermal Power Plants

EPRI RP 3657-01

Final Report, September 1995

Prepared by
CE Holt Co.
201 S. Lake Avenue
Pasadena, CA 91101

Principal Investigators:

John Brugman
Mai Hattar
Kenneth Nichols
Yuri Esaki

Prepared for
Electric Power Research Institute
3412 Hillview Avenue
Palo Alto, California 94304

EPRI Project Manager
Dr. Evan Hughes

DISCLAIMER OF WARRANTIES AND LIMITATION OF LIABILITIES

ORGANIZATION(S) THAT PREPARED THIS REPORT:

CE HOLT CO.

CONTENTS

Section **Page**

1 **Executive Summary** 1-1

2 **Introduction** 2-1
 Project Goal 2-1
 Project Description 2-1

3 **Background** 3-1
 Problem Statement 3-1
 Previous EPRI Work by Holt 3-2

4 **Method of Approach** 4-1
 Introduction 4-1
 Terminology 4-2
 General Methodology 4-2
 Assumptions 4-8

5 **Baseline Technologies** 5-1
 Introduction 5-1
 Air-cooled Commercial Binary 5-1
 Water-cooled Commercial Binary 5-7
 Commercial Dual Flash 5-12
 Conclusions 5-25

6 **Advanced Binary Cycles** 6-1
 Mixed Fluid Binary: Hydrocarbon Mixtures 6-1
 Synchronous Speed Turbine 6-13
 Metastable Expansion 6-15
 Binary Cycle: Hot Dry Rocks 6-22

7 **Advanced Flash Cycles** 7-1
 Dual Flash/Rotary Separator Turbine 7-1

Dual Flash/Steam Reheater 7-6
Sub-atmospheric Flash 7-10
Dual Flash/Hot Water Turbine 7-12

8 **Hybrid Cycles** 8-1
Flash/Binary Bottoming Cycles 8-1
Dual Flash/Back-Pressure-Turbine Binary 8-5
Dual Flash/Gas Turbine Hybrid 8-11
Binary/Gas Turbine Hybrid 8-15

9 **Environmental Impact** 9-1
Introduction 9-1
Technology Independent Impacts 9-1
Technology Dependent Impacts 9-3
Rankings 9-6

10 **Next Generation Geothermal Power Plant Technology** 10-1
Concept Ranking Methodology 10-1
Levelized Busbar Cost Methodology 10-2
Levelized Busbar Cost Results 10-6
Concept Ranking Results 10-7

11 **Conclusions and Recommendations** 11-1
Air-cooled Commercial Binary 11-2
Mixed Fluids 11-2
Synchronous Turbine 11-2
Metastable Expansion 11-3
Rotary Separator Turbine 11-3
Reheater 11-3
Sub-atmospheric Flash 11-3
Hot Water Turbine 11-4
Dual Flash/Binary Bottoming 11-4
DF Back-Pressure-Turbine 11-4
Hot Dry Rock 11-4
Gas Turbine Hybrid Cycles 11-5

12 **References** 12-1

Appendix A
Gathering System Spreadsheet A-1
Well Pump Spreadsheet A-1
Parasitics A-2
Heat Exchanger Calculations A-2

Appendix B - Synchronous Turbine Generators B-1
Introduction B-1
Technical Background B-1
Thermodynamic Database B-4
Baseline Case - Vale B-5
Performance B-13
Cost B-14
Technical Risks B-16

FIGURES

Figures | Page

5-1 Air-cooled Commercial Binary Cycle: Process Flow Diagram......5-2
5-2 Air cooled Commercial Binary: Specific Output Curves 5-4
5-3 Air-cooled Commercial Binary Cycle: Specific Output v. Resource Temperature 5-5
5-4 Air-cooled Commercial Binary Cycle: Specific Capital Cost v. Resource Temperature 5-7
5-5 Water-cooled Commercial Binary Cycle: Process Flow Diagram...5-9
5-6 Specific Capital Cost v. Condenser Pinch, Vale,..........................5-10
5-7 Water-cooled Binary: Specific Output v. Turbine Inlet Pressure...5-11
5-8 Water-cooled Binary: Specific Cost v. Turbine Inlet Pressure.......5-11
5-9 Commercial Dual Flash Cycle: Process Flow Diagram 5-14
5-10 Condenser Approach Optimization, Vale.......................................5-18
5-11 Comparison of Vacuum Systems 5-19
5-12 Cooling Water Circulation Optimization, Vale 5-20
5-13 Commercial Dual Flash Cycle: Economic Optimum Specific Output v. Resource Temperature...5-21
5-14 Commercial Dual Flash Cycle: Specific Capital Cost v. Resource Temperature 5-22
5-15 Baseline Technologies - Specific Output v. Resource Temperature 5-25
5-16 Baseline Technologies - Specific Capital Cost v. Resource Temperature 5-26
6-1 Mixed Fluids Specific Output Curves, Surprise Valley 6-5
6-2 Mixed Fluids Specific Capital Cost Curves, Surprise Valley..........6-5
6-3 Specific Capital Cost v. Cold End Pinch, Surprise Valley 6-6
6-4 Mixed Fluids Specific Output Curve, Vale.......................................6-7
6-5 Mixed Fluids Specific Capital Cost Curves, Vale 6-8
6-6 Mixed Fluids Specific Output Curves, Raft River 6-9
6-7 Mixed Fluids Specific Capital Cost Curves, Raft River..................6-10

6-8 Mixed Fluids Optimizing Condensing Temperatures, Thermo Hot Springs 6-11
6-9 Mixed Fluids Specific Output Curves, Thermo Hot Springs 6-12
6-10 Mixed Fluids Specific Capital Cost Curves, Thermo Hot Springs . 6-13
6-11 Metastable Expansion for Hypothetical Cycle 6-16
6-12 Comparison of NIST-12 and Starling Equation of State 6-17
6-13 Ratio of Metastable Expansion and Commercial Binary Costs 6-18
6-14 Metastable Expansion Binary Specific Output Curves, Vale 6-20
6-15 Metastable Expansion Binary Specific Output Curves, Surprise Valley 6-21
6-16 Metastable Expansion Binary Specific Output Curves, Raft River 6-21
6-17 Hot Dry Rocks Surface System Design 6-22
7-1 Dual Flash/Rotary Separator Turbine: Process Flow Diagram 7-2
7-2 RST: Incremental Change in Specific Output v. Commercial Dual Flash 7-4
7-3 RST: Specific Capital Cost v. Flash Pressure, Salton Sea 7-5
7-4 Dual Flash/Steam Reheater: Process Flow Diagram 7-7
7-5 Dual Flash/Steam Reheater: Incremental Change in Specific Power Output 7-8
7-6 Sub-atmospheric Flash Cycle: Specific Output Comparison 7-11
7-7 Sub-atmospheric Flash Cycle: Specific Cost Comparison 7-12
7-8 Dual Flash/Hot Water Turbine: Process Flow Diagram 7-13
7-9 Brine Utilization Comparison 7-15
7-10 Specific Plant Cost Comparison 7-16
8-1 Dual Flash/Binary Bottoming Cycle: Process Flow Diagram 8-2
8-2 Single Flash/Binary Bottoming Cycle: Process Flow Diagram 8-2
8-3 Dual Flash/Back-Pressure-Turbine Binary: Process Flow Diagram 8-6
8-4 DFBPT: Flash Pressure Optimization, Glass Mountain 8-8
8-5 DFBPT: Specific Plant Cost v. LP Flash Pressure, Glass Mountain 8-10
8-6 Dual Flash/Gas Turbine Hybrid I: Process Flow Diagram 8-12
8-7 Dual Flash/Gas Turbine Hybrid II: Process Flow Diagram 8-12
8-8 Dual Flash/Gas Turbine Hybrid - Specific Output v. Resource 8-15
8-9 Gas Turbine - Binary Hybrid I: Process Flow Diagram 8-16
8-10 Gas Turbine - Binary Hybrid II: Process Flow Diagram 8-17
8-11 Temperature-Enthalpy Diagram: Heat Exchangers in Series 8-18
8-12 Temperature-Enthalpy Diagram: Heat Exchangers in Parallel 8-18

B-1 Optimized Performance Chart, Axial Flow Turbines B-3
B-2 Multi-Stage, Radial-Inflow Turbine B-7
B-3 Back-to-Back Axial Flow Turbines B-9
B-4 Axial Flow Turbine: Typical Dimensions B-11

B-5 Axial Flow Turbine Detail..B-12

TABLES

Tables		Page
4-1	EPRI Supplied Resource Data	4-3
4-2	Power Plant Multipliers	4-6
4-3	Meteorological Data	4-7
4-4	Wellhead Pressures for Self-Flowing Resources	4-9
4-5	Noncondensable Gas Content (NCG) of Geothermal Brine	4-9
5-1	Air-cooled Commercial Binary: Summary of Cases	5-6
5-2	Air-cooled Commercial Binary Cycle: End of Reservoir Results	5-8
5-3	Turbine Inlet Conditions for Optimum Cases	5-12
5-4	Comparison of Water-cooled and Air-cooled Binary Cycles	5-13
5-5	Commercial Dual Flash Plant Process Description	5-17
5-6	Commercial Dual Flash Plant Process Parameter Summary	5-17
5-7	Commercial Dual Flash Cycle: Major Equipment and Total Plant Costs	5-23
6-1	Effect of Varying Heat Transfer Coefficient on Specific Plant Cost	6-2
6-2	Comparison of Mixed Fluids Cycle with Air-cooled Commercial Binary Cycle	6-3
6-3	Mixed Fluids Equipment and Total Plant Cost Summary	6-4
6-4	Surprise Valley Mixed Fluids Case Summary	6-4
6-5	Vale Mixed Fluids Case Summary	6-7
6-6	Raft River Mixed Fluids Case Summary	6-9
6-7	Thermo Hot Springs Mixed Fluids Case Summary	6-11
6-8	Comparison of Turbines: Specific Cost and Output	6-14
6-9	Metastable Expansion: Comparison of Equipment Costs	6-19
6-10	Metastable Expansion: Summary of Cases	6-22
7-1	Dual Flash/RST Cycle Process Summary	7-4
7-2	Dual Flash/Steam Reheater Process Parameters	7-9
7-3	Dual Flash/Steam Reheater Process Parameters and Specific Capital Cost	7-9
7-4	Sub-atmospheric Flash Cycles: Specific Capital	

	Cost and Output	7-14
8-1	Summary of Flash/Binary Bottoming Cycle Cases	8-4
8-2	Flash/Binary Bottoming Comparison Summary	8-5
8-3	Dual Flash/Back-Pressure-Turbine Binary Hybrid Performance Summary	8-9
8-4	Dual Flash/Back-Pressure-Turbine Binary - Major Equipment and Total Plant Costs	8-10
8-5	Dual Flash/Gas Turbine Hybrid - Evaluation of Cycle Alternatives	8-14
8-6	Specific Output, kWh/ 1000 lb Brine	8-19
8-7	Exhaust Gas Heat Input	8-20
8-8	Specific Capital Costs	8-20
9-1	Gas Turbine Exhaust	9-5
9-2	Environmental Impact Rankings	9-6
10-1	Specific Output, kWh/1000 lb Brine	10-2
10-2	Levelized Busbar Cost of Power Generation Assumptions	10-3
10-3	Specific Cost, $/kW	10-4
10-4	Levelized Busbar Cost of Power Plant Assumptions	10-5
10-5	Operating and Maintenance Cost Assumptions	10-5
10-6	Levelized Revenue Requirement, ¢/kWh	10-6
10-7	Levelized Revenue Requirement for Natural Gas Cycles, ¢/kWh	10-8
10-8	Concept Ranking	10-8
B-1	Data Base Differences	B-6
B-2	Radial Flow Turbine Parameters for One of Six Parallel Units	B-10
B-3	Axial Flow Turbine Parameters for One of Four Parallel Units	B-11
B-4	Turbine Staging	B-13
B-5	BN Power Output	B-13
B-6	BH Power Output	B-14
B-7	Power Unit Cost	B-15
B-8	Cost of Stages	B-16
B-9	Power Unit Cost	B-16

1

EXECUTIVE SUMMARY

A number of current and prospective power plant concepts were investigated to evaluate their potential to serve as the basis of the next generation geothermal power plant (NGGPP). The NGGPP has been envisaged as a power plant that would be more cost-competitive (than current geothermal power plants) with fossil fuel power plants, would efficiently use resources and mitigate the risk of reservoir under-performance, and minimize or eliminate emission of pollutants and consumption of surface and ground water.

Power plant concepts were analyzed using resource characteristics at ten different geothermal sites located in the western United States. Concepts were developed into viable power plant processes, capital costs were estimated and levelized busbar costs determined. Thus, the study results should be considered as useful indicators of the commercial viability of the various power plants concepts that were investigated.

Broadly, the different power plant concepts that were analyzed in this study fall into the following categories: commercial binary and flash plants, advanced binary plants, advanced flash plants, flash/binary hybrid plants, and fossil/geothermal hybrid plants. Commercial binary plants were evaluated using commercial isobutane as a working fluid; both air-cooling and water-cooling were considered. Advanced binary concepts included cycles using synchronous turbine-generators, cycles with metastable expansion, and cycles utilizing mixtures as working fluids.

Dual flash steam plants were used as the model for the commercial flash cycle. The following advanced flash concepts were examined: dual flash with rotary separator turbine, dual flash with steam reheater, dual flash with hot water turbine, and sub-atmospheric flash. Both dual flash and binary cycles were combined with other cycles to develop a number of hybrid cycles: dual flash binary bottoming cycle, dual flash backpressure turbine binary cycle, dual flash gas turbine cycle, and binary gas turbine cycle.

Results of this study indicate that dual flash type plants are preferred at resources with temperatures above 400°F. Closed loop (binary type) plants are preferred at resources with temperatures below 400°F. A rotary separator turbine upstream of a dual flash plant can be beneficial at Salton Sea, the hottest resource, or at high temperature resources where there is a significant variance in wellhead pressures from well to well. Full scale demonstration is required to verify cost and performance.

Hot water turbines that recover energy from the spent brine in a dual flash cycle improve that cycle's brine efficiency. Prototype field tests of this technology have established its technical feasibility.

If natural gas prices remain low, a combustion turbine/binary hybrid is an economic option for the lowest temperature sites.

The use of mixed fluids appear to be an attractive low risk option. The synchronous turbine option as prepared by Barber-Nichols is attractive but requires a pilot test to prove cost and performance. Dual flash binary bottoming cycles appear promising provided that scaling of the brine/working fluid exchangers is controllable.

Metastable expansion, reheater, sub-atmospheric flash, dual flash backpressure turbine, and hot dry rock concepts do not seem to offer any cost advantage over the baseline technologies.

If implemented, the next generation geothermal power plant concept may improve brine utilization but is unlikely to reduce the cost of power generation by much more than 10%. Colder resources will benefit more from the development of a next generation geothermal power plant than will hotter resources.

All values presented in this study for plant cost and for busbar cost of power are relative numbers intended to allow an objective and meaningful comparison of technologies. The goal of this study is to assess various technologies on an common basis and, secondarily, to give an approximate idea of the current costs of the technologies at actual resource sites. Absolute costs at a given site will be determined by the specifics of a given project.

2

INTRODUCTION

Project Goal

The goal of this project is to develop concepts for the next generation geothermal power plant(s) (NGGPP). These plants, compared to existing plants, will generate power for a lower levelized cost and will be more competitive with fossil fuel fired power plants. The NGGPP will utilize geothermal resources more efficiently and will be designed to mitigate the risk of reservoir under-performance. The NGGPP design will attempt to minimize emission of pollutants and consumption of surface water and/or geothermal fluids for cooling service.

Project Description

This project was funded by a consortium of governmental agencies and Western utilities. The Electric Power Research Institute (EPRI) acted as manager on behalf of this consortium. CE Holt Company (henceforth Holt) is the prime contractor for the project, and Fuji Electric Company and Barber-Nichols are sub-contractors on the project. John Brugman and Mai Hattar of Holt, Kenneth Nichols of Barber-Nichols Inc. and Yuri Esaki of Fuji Electric Company, Ltd. are the principal investigators.

Geothermal sites were chosen such that the NGGPP concepts would be evaluated using "real world" resources, and would include the effect of noncondensable gas content, solids composition, resource temperature, well costs, ambient conditions, etc. on each concept's performance.

The project was funded in November, 1993 and Holt commenced work in December 1993. A steering committee comprised of representatives of the Department of Energy, Southern California Edison, Bonneville Power Administration, and EPRI directed the project. The steering committee held two review meetings to evaluate the progress of the project; the first on May 25, 1994, and the second on November 2 and 3, 1994. The second review meeting was also attended by a representative from Biphase Energy for the session on the rotary separator turbine, and by a representative from Fuji Electric Company. The Phase I study was completed in December, 1994.

3

BACKGROUND

Problem Statement

Geothermal energy is an indigenous, sustainable and environmentally benign energy source. Potentially, it "could supply the energy needs of man for a millenia" (DiPippo, 1980). Geothermal power plants harness geothermal energy to produce electricity, and are an attractive source of power since they usually emit minimal combustion products into the atmosphere compared to fossil fuel plants.

Presently, several technologies are being used to convert geothermal energy into electric power, and many other technologies are in various stages of development. Technologies currently in use and development will likely define the next generation geothermal power plant(s) (NGGPP) which will be designed to accommodate changes in geothermal resource characteristics and demands of the marketplace. This study evaluates various concepts that may form the basis for developing the NGGPP.

The NGGPP, as defined, "will have higher power conversion efficiencies, better geothermal resource utilization, and lower power generation costs than current proven geothermal power generation technologies" (EPRI RFP3657-01, 1993). The NGGPP performance will be relatively unaffected by risks associated with geothermal reservoir under-performance. The NGGPP is perceived to have a minimal impact on the environment and resources. The NGGPP will minimize cooling water requirements and maximize injection of geothermal fluids to enhance resource utilization and it will minimize or eliminate emission of greenhouse gases and acid gases.

Currently, dry steam and hydrothermal resources have been exploited commercially. The NGGPP design will be flexible in that it can be adapted to utilize hot dry rock and geopressured resources. Moreover, in view of the fact that a number of resources are low temperature, the ideal NGGPP should be able to efficiently generate power from resources below 300°F. In order to standardize design and minimize construction costs, the NGGPP design should be modular to the extent possible.

Previous EPRI Work by Holt

In 1979, at EPRI's "Third Annual Geothermal Conference and Workshop", the workshop topic was "Looking Ahead to the Next Generation of Geothermal Power Systems". The workshop focused on three areas of development: (1) desired characteristics of future systems, (2) emerging technology and (3) research and development needs. Several promising technologies that could impact future geothermal power plants were identified by workshop participants. The list of technologies included well stimulation techniques, downwell pumps, advanced flash systems, advanced binary systems, and two-phase expanders. Details of the workshop can be found in the conference proceedings (EPRI, 1979).

From 1979 to 1983 EPRI funded the "Assessment of Advanced Geothermal Energy Conversion Concepts" program under EPRI RP 1673-1. As a part of this program a number of advanced geothermal concepts were investigated, including advanced flash steam cycles, advanced binary cycles, hybrid cycles (fossil/geothermal and solar/geothermal), and total flow devices (rotary separator turbine, helical expander, bladeless turbine, LLL impulse turbine, LLL reaction turbine, Elliot turbine and Horst expander). Partial results of this program were presented at the "Fifth Annual Geothermal Conference and Workshop" (Holt et al, 1981).

Under EPRI contract RP1673-4, Holt evaluated methane and hot water hybrid cycles. Hybrid cycles were calculated to produce as much as 30% more electricity than if methane was used separately in a fossil fuel power plant and hot water in a binary cycle plant. This 30% improvement is based on a cycle in which enough methane is combusted so that exhaust heat provides all necessary energy to vaporize the working fluid.

Under EPRI contract RP1671-5 and DOE contract DE-AC07-85ID12578, Holt designed, built and successfully operated a hybrid cycle power plant on the Pleasant Bayou geopressured resource in Texas. Since that geopressured resource contains both methane and geothermal brine, a hybrid cycle was employed to fully utilize the resource. In the hybrid cycle at Pleasant Bayou, gas was burned in engines to generate electricity directly. Exhaust heat from the engines was then combined with heat from the brine to generate additional electricity in a binary cycle.

Holt performed work under EPRI project, RP1671-07, to extend the application of hybrid plants to geothermal resources which are not geopressured and to include biomass fuels as an alternative to natural gas.

4

METHOD OF APPROACH

Introduction

The development of the next generation geothermal power plant entails choosing realizable technologies from a host of power plant concepts. In consultation with the steering committee fifteen specific geothermal power plant concepts were identified for evaluation in this study. These are listed below. The first three concepts represent proven technology and provide a baseline for evaluating other technologies. The remaining concepts are in various stages of development and will require demonstration on a significant scale to be classified as commercial.

Baseline Technologies

- Commercial air-cooled binary
- Commercial water-cooled binary
- Commercial dual flash

Advanced Binary Cycles

- Mixed fluids
- Synchronous speed turbine
- Metastable expansion
- Hot dry rocks

Advanced Flash Cycles

- Dual flash/rotary separator turbine
- Dual flash/steam reheater
- Sub-atmospheric flash
- Dual flash/hot water turbine

Hybrid Cycles

- Dual flash/binary bottoming cycle
- Dual flash backpressure turbine/binary
- Dual flash/gas turbine
- Binary cycle/gas turbine

These concepts were applied to existing, known geothermal sites in order to base the study on real-world applications. Geothermal sites were chosen so that the NGGPP technologies would be applicable to resources with widely varying characteristics. Detailed resource characteristics for the selected sites are listed in Table 4-1 (provided by EPRI) which shows that the selected resources cover a broad range of temperatures, noncondensable gas composition, and solids content. The selected resources include Clear Lake-Geysers which is a hot dry rock resource. The following resources were included (resource temperature is shown in parenthesis):

- Clear Lake-Geysers, California (375°F)
- Coso Hot Springs, California (525°F)
- Desert Peak, Nevada (425°F)
- Dixie Valley, Nevada (450°F)
- Glass Mountain, California (510°F)
- Raft River, Idaho (300°F)
- Salton Sea, California (570°F)
- Surprise Valley, California (375°F)
- Thermo Hot Springs, Utah (265°F)
- Vale, Oregon (330°F)

Terminology

Throughout this report the results of the evaluation of NGGPP technologies are presented in terms of specific output and specific capital cost. Specific output provides an index for the thermodynamic performance of power plant cycles and is sometimes referred to as brine utilization rate. It is defined as:

$$\text{Specific output} = \text{Net power (kW)/Brine mass flow rate (1000 lb/hr)} \qquad (4\text{-}1)$$

Specific capital cost is an index used for assessing the relative cost of power plant technologies and is defined as:

$$\text{Specific capital cost} = \text{Total plant cost (\$)/Net power (kW)} \qquad (4\text{-}2)$$

Total plant cost in equation 4-2 is the capital expense required to develop the wellfield and build the power plant but does not include the cost of land, interest on capital and owner's project expenses. All costs in this study were calculated in 1994 dollars.

General Methodology

Cycle concepts were optimized for each applicable site using minimum specific capital cost as the figure of merit. Cycle optimization to minimize specific capital cost was accomplished in two stages: (1) performance analysis and (2) economic

Table 4-1
EPRI Supplied Resource Data

Resource	Glass Mountain, California	Salton Sea, California	Surprise Valley, California	Thermo Hot Springs, Utah	Vale, Oregon
Geothermal Fluid Temperatures, °F		480 - 660	338 - 428	262 - 268	320 - 340
Initial	510	570	375	265	330
Final	430	480	325	235	290
Average Well Flowrate, kph	300	1,300	500(1)	300(1) est.	300(1) est.
TDS, ppm	3800 flashed	150,000-250,000	800 - 1,200	1,500	600 - 1,200
Average Well Depth, ft	7,500	6,000	5,000	5,000	5,000
Reservoir Pressure					
Type	underpressured hydrostatic	hydrostatic	hydrostatic	hydrostatic	hydrostatic
At Datum		0.414 psi/ft pressure gradient			
Average Well Cost, $MM					
Producer	1.5	2.5	1.5	1.5	1.5
Injector					
Percent Replacement Wells Required, %/yr	unknown				
Minimum Reinjection Temperature, °F					
Relative Corrosivity	low	high	very low	low	very low
Scaling Potential	low	high	low	low	low
Noncondensable Gas Content, w %	0.17	0.014	> 0.2	> 0.2	> 0.2
Percent Dry Holes	est. 25	none to date	20 - 25	20 est.	unknown
Reservoir Can Be Dispatched	unknown	no	possible	possible	possible
Potential Energy 30 years, MWe					
open cycle	=>500	=>500 (onshore)	350	50*	=>350
closed cycle	=>500	=>500 (onshore)	500	50*	=>500
USGS 790 est.	750(2)	3400	500(2)	(3)	870
Water availability	no	possible	possible	no	possible

(1) 700,000 kph when pumped
(2) by Ebasco
(3) based on USGS 790 assuming 33% efficiency

Table 4-1
EPRI Supplied Resource Data (continued)

Resource	Clear Lake - Geysers	Coso Hot Springs, CA	Desert Peak Nevada	Dixie Valley Nevada	Raft River Idaho
Geothermal Fluid Temperatures, °F	N/A	392 - 653	400 - 425	400- 480	302
Initial	375	525	425	450	300
Final	330	445	365	380	270
Average Well, Flowrate, kph	200 on pump	400	500	1,000	190
TDS, ppm	>4000	3,700 - 8,000	6,700	1,300	1,500 - 6,700 mg/l
Average Well Depth, ft	8,000	6,000	6,000	10,000	5,000
Reservoir Pressure					
Type	4000 psi inject 1500 psi produce	hydrostatic w/steam cap	hydrostatic	hydrostatic	hydrostatic
At Datum		initial 850 psia @ 2,950 ft.	N/A	initial 3200 psia @ 8,450 ft.	N/A
Average Well Cost, $MM					
Producer	6.0 includes	1.5	1.5	1.5	1.5
Injector	2 for hyd. fracturing trt.				
Percent Replacement Wells Required, %/yr	6	2	2	2	2
Minimum Reinjection Temperature, °F	140	220	185	185	150
Relative Corrosivity	low	low	moderate	low	low
Scaling Potential	low	high	moderate	low	low
Non-condensable[1] Gas Content, wt %	0	1,000 - 25,000 mg/kg	0.029	0.20	est. < 0.2
Percent Dry Holes	0	25 during exploration	17 during exploration	30 during exploration	Estimated 20
Reservoir Can Be Dispatched	possible	not currently	not currently	not currently	possible
Potential Energy 30 years MWe					
open cycle	N/A	160	200	75	0
closed cycle	975	240	200	100	25
USGS 790 est.	975	650	750		100
Water availability	possible	no	no	no	no

(1) 700,000 kph when pumped
(2) by Ebasco
(3) based on USGS 790 assuming 33% efficiency

analysis. In the performance analysis stage a number of process cases were simulated for a given cycle and resource to obtain a set of cases that could be expected to contain the economic optimum cycle. The thermodynamic optimum for the subject cycle was determined at this time. Identification of the optimum cycle was completed in the economic analysis stage wherein cost evaluation of the cases generated during performance analysis was performed. A discussion of performance and economic analysis is presented below.

Rankings for NGGPP concepts were developed using levelized busbar costs as the primary criterion. The methodology for calculating levelized busbar costs is presented along with NGGPP concept rankings in Section 10.

Performance Analysis: Concept Development and Cycle Analyses

Performance analysis of each concept involved first developing the process design by preparing a process flow diagram and a completed process model. The process model was used to simulate cycle performance by varying the process variables, and several cases were developed corresponding to different sets of process variables for each cycle. Process variables were chosen to obtain the cycle specific output over a range of important process constraints such as pressures, temperatures, pinch points, temperature approaches, etc. The range of process constraints was based on Holt's experience in designing and building commercial geothermal power plants.

The NGGPP concepts examined in this study included variations of commercial binary and/or commercial flash cycles. Holt has proprietary mathematical models for designing both types of cycles, and these Holt models were used for the process design of most cycles. Details of the Holt models for the commercial binary and commercial flash cycles are presented in the sections on the air-cooled commercial binary and commercial dual flash cycles, respectively.

Economic Analysis: NGGPP Capital Cost Evaluation

For the purpose of economic optimization, the optimum plant design was defined as the plant design corresponding to minimum specific capital cost. In order to compare all concepts on an even basis, a power plant design with a nominal capacity of 50 MW (net) was used as the basis for estimating capital costs for all concepts. With this basis, the objective of minimizing specific capital cost ($/kW) was reduced to the objective of minimizing total plant cost. Throughout this report minimum specific capital cost has been used as the figure of merit for identifying the economic optimum. Since variable costs represent a relatively small fraction of the levelized busbar costs, the fixed costs, which are principally capital costs, determine the overall economics. Thus, the specific capital cost is as good predictor of the ultimate levelized busbar costs.

Total plant cost was calculated as the sum of power plant cost and wellfield cost (including gathering system). Power plant costs were estimated from major equipment costs using the installation factor method: major equipment costs were summed and multiplied by a plant cost multiplier to account for engineering and installation costs. The multipliers used in this study are based on Holt experience in designing and building both binary and flash geothermal power plants in the recent past, and are listed in Table 4-2. Wellfield costs were calculated using production and injection well costs provided by EPRI and gathering system costs were calculated by Holt. Production and injection well costs for the various geothermal sites are listed in Table 4-1. Using total plant costs, specific capital costs were obtained for each of the designs developed in the performance analysis stage, and the cycle design corresponding to minimum specific capital cost was chosen as the economically optimum cycle design.

Table 4-2
Power Plant Multipliers

	Installed Cost Multiplier	
Site	Binary	Flash/Flash Derivatives
Clear Lake-Geysers, California	2.8	2.53
Coso Hot Springs, California	2.8	2.53
Desert Peak, Nevada	2.8	2.53
Dixie Valley, Nevada	2.8	2.53
Raft River, Idaho	2.96	2.53
Glass Mountain, California	2.8	2.53
Salton Sea, California	2.8	2.53
Surprise Valley, California	2.8	2.53
Thermo Hot Springs, Utah	3.1	2.53
Vale, Oregon	2.83	2.53

The details of the cost estimation methods are presented in Appendix A. Major equipment costs were obtained from Holt's extensive database supplemented where necessary by current vendor quotations. For similar pieces of equipment, costs from the same vendor were used for evaluating all competing technologies. For example, turbine costs for all evaluated NGGPP concepts are based on turbine cost data from Fuji Electric Company, a co-investigator in this study. The underlying philosophy has been to evaluate competing technologies within the same frame of reference. The choice of vendors was based on the experience and reputation of vendors supplying geothermal power plant installations in the United States. Fuji Electric Company's turbines, for example, account for approximately 40% of the geothermal power plant capacity in the United States.

Wet and dry bulb temperatures (Table 4-3) were obtained from the Department of the Navy publication, "Facility Design and Planning - Engineering Weather Data" (Department of the Navy, 1978). This study used annual average weather conditions resulting in a constant (not seasonally differentiated) cost of power.

Table 4-3
Meteorological Data

Site	Avg Dry Bulb Temp (°F)	Avg Wet Bulb Temp (°F)	Altitude (ft)
Clear Lake-Geysers, California	59	51	1,630
Coso Hot Springs, California	60	49	4,200
Desert Peak, Nevada	50	39	4,600
Dixie Valley, Nevada	50	39	3,500
Raft River, Idaho	47	38	4,500
Glass Mountain, California	53	46	5,000
Salton Sea, California	74	56	-100
Surprise Valley, California	50	39	4,600
Thermo Hot Springs, Utah	49	39	5,800
Vale, Oregon	51	41	4,100

Environmental Impact

The environmental impact of each NGGPP concept was assessed. Among the impacts evaluated were surface and ground water use and pollutant emissions for each optimized cycle for each resource.

A liquid redox sulfur plant for H_2S is included where required, and it is assumed that carbon dioxide, a greenhouse gas, can be emitted to the atmosphere.

Next Generation Geothermal Power Plant Concepts Ranking

This study ranks concepts primarily based on busbar costs. Second level considerations are used to discriminate between technologies only in cases where two or more concepts have almost equal busbar costs. The following second level considerations are used to "break ties":

- Specific Output (Brine utilization efficiency)
- Environmental considerations

Assumptions

This section presents the assumptions that are common to all NGGPP concepts evaluated in this study. As mentioned in the previous section, the characteristics of each site and meteorological data for each resource are presented in Table 4-1 and 4-3, respectively. In addition, Table 4-4 provides the maximum wellhead pressure for self-flowing wells, and Table 4-5 provides the composition of noncondensable gases in the brine for each resource. These tables are used to obtain the following data for each site as required for each concept:

- Average dry/wet bulb temperatures
- Composition of noncondensable gases in the geothermal brine
- Site altitude
- Geothermal fluid temperature
- Minimum injection temperature
- Average well flowrates
- Average well cost
- Concentration of total dissolved solids
- Probability of a dry hole
- Maximum flash pressure

The following general assumptions have been used as applicable in the development of performance and cost models for the various NGGPP concepts:

- Resources with temperatures below 400°F are assumed to have pumped wells; resources with temperatures above 400°F are assumed to be free flowing.

Table 4-4
Wellhead Pressures for Self-Flowing Resources

Resource	Temperature (°F)	Pressure (psia)
Coso	525	130
Desert Peak	425	90
Dixie Valley	450	100
Glass Mtn.	510	151
Salton Sea	570	325

Table 4-5
Noncondensable Gas Content (NCG) of Geothermal Brine

Site	NCG wt ppm	H_2S wt ppm
Clear Lake-Geysers, California	0	0
Coso Hot Springs, California	20,000	105
Desert Peak, Nevada	290	4
Dixie Valley, Nevada	2,000	5
Raft River, Idaho	2,000	0.2
Glass Mountain, California	1,700	20
Salton Sea, California	1,360	16
Surprise Valley, California	2,000	20
Thermo Hot Springs, Utah	2,000	20
Vale, Oregon	2,000	20

- For pumped wells discharge pressure at the surface discharge flange is the maximum of: (a) Hot brine vapor pressure + 50 psi or (b) 143 psia
- There are no elevation changes between the plant site and the pumped well locations.

- There are no elevation changes between the plant site and the injection well locations.
- Well drawdown is based on a factor of 2,500 lb/(hr-psi). Submerged pumps are placed 150 feet under the calculated draw down levels.
- Gathering system costs are based on an estimated configuration and layout of the gathering system. Appendix A contains the gathering system calculation details.
- Production well drilling costs are adjusted according to the probabilities of hitting a dry hole listed in Table 4-1 for each resource, as follows: The required number of production wells is calculated by dividing the required brine flow rate by the average well flow rate listed in Table 4-1. The required number of production wells is then divided by one minus the fraction of dry holes to yield the total number of production wells that would have to be drilled. The fraction of dry holes is based on exploratory, not production, drilling. (Total plant cost could be lowered if the percentage of dry holes is found to be lower during production well drilling.
- Injection pump discharge pressure is 100 psi plus the vapor pressure of the outlet brine.
- Injection pumps are sized to be 500 hp maximum. A minimum of two 50% pumps are used. Whenever the pumping power exceeds the 500 hp limitation, another pump is added to the injection system.
- Pumped wells have an average flow of 700,000 lb/h, and a maximum flow of 725,000 lb/h. Whenever the average well flow exceeds 725,000 lb/h, more wells are added to the field until the resulting well flow rate is below 725,000 lb/h.
- All heat exchanger costs are adjusted for pressure so that different pressure levels can be compared on the same basis.
- If required, the plant cost for open cycles includes the cost of a sulfur plant for H_2S abatement; H_2S emissions are limited to 100 g/MWh or 11 lb/h for a 50 MW plant.

5

BASELINE TECHNOLOGIES

Introduction

Flash and binary cycles are the predominant technologies used for generation of geothermal electricity (Fridleifsson and Freeston, 1994). These two technologies were chosen to serve as baseline technologies in this study since they are proven technologies that find widespread commercial use and are well optimized. Throughout this report the terms commercial flash and commercial binary will represent the baseline flash and binary cycles, respectively. Both air-cooled commercial binary and water-cooled commercial binary cycles have been considered.

The commercial binary cycle is a Rankine cycle which uses a hydrocarbon as the working fluid, and is sometimes referred to as an organic Rankine cycle. This technology has been applied successfully to resources below 350°F. The majority of binary cycle installations are air-cooled commercial binary cycles which rely on air-cooling for condensation of the working fluid. For example, power plants at Steamboat Springs, Nevada, and Mammoth, California are air-cooled commercial binary plants. Installations where cooling water is available utilize the water-cooled commercial binary cycles. For example, the Second Imperial Geothermal Company (SIGC) power plant at Heber is a water-cooled commercial binary plant.

A discussion of the air-cooled commercial binary cycle is presented in the following section; the next section discusses the water-cooled commercial binary cycle.

Air-cooled Commercial Binary

Cycle Process Flow

Figure 5-1 is the process flow diagram of the air-cooled commercial binary cycle. Hot geothermal fluid (brine) is used to heat and vaporize a working fluid in a series of heat exchangers. The hot working fluid vapor is then expanded through a turbine to generate electricity. The turbine exhaust is subsequently condensed in an air-cooler. The cycle is complete when the condensed working fluid is pumped back to be heated by the hot brine.

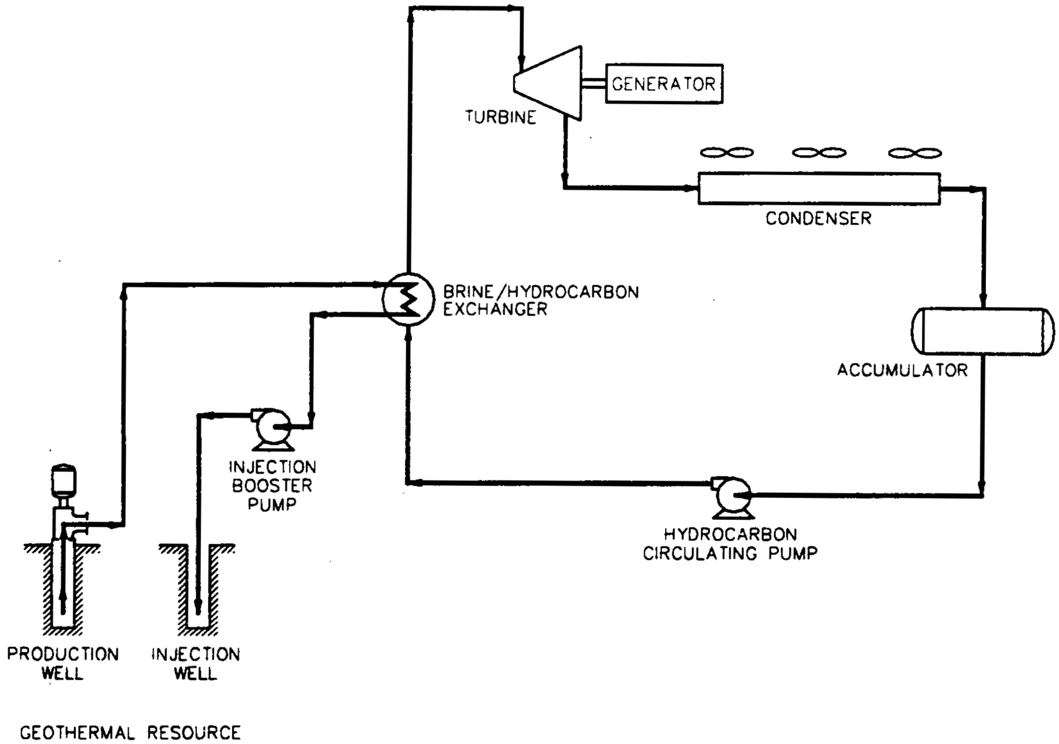

Figure 5-1
Air-cooled Commercial Binary Cycle: Process Flow Diagram

Binary cycles in this study were modeled using commercial isobutane as the working fluid. This is a common working fluid in binary cycle power plants. The working fluid was assumed to be composed of 96.06 mole percent isobutane, 1.77 mole percent normal butane, and 2.17 mole percent propane.

Performance Analysis

Description. As noted in Section 4 the optimum cycle was identified by performing process calculations for several cases for each plant site. For the binary cycles these calculations were performed using a proprietary Holt model that incorporates the BWR/Starling equation of state. This model computes thermodynamic state points, power output, brine flow requirements, and parasitic loads for binary cycles, and was used to calculate these quantities for a 50 MW (net) power plant design. The results of the model calculations were used to obtain specific output for various process cases.

The calculations for final (end of run) resource conditions were performed assuming constant turbine inlet volumetric flow rate. Turbine inlet mass flow rate and pressure were adjusted to maintain a constant volumetric flow rate, and the turbine efficiency was adjusted to account for operating at off-design conditions.

Assumptions. Modeling assumptions used to evaluate the performance of all the concepts are listed in Section 4. For example, basic resource and weather data used for these calculations are listed in Table 4-1 and 4-3. The following assumptions apply specifically to binary plant performance calculations:

- Commercial isobutane is the working fluid.
- Turbine adiabatic efficiency: 85.0%
- Working fluid pump efficiency: 80%.
- Generator and gear efficiency: 94.0%
- Hydrocarbon loop pressure drop: 50 psi (includes pressure drop for piping, control valve, and heat exchangers)
- The air condenser was sized to give a 34°F approach on the cold end with a 14.8°F pinch unless a greater pinch was required to avoid a temperature cross in the heat exchanger. Pinch is defined as the minimum temperature difference between the hot and cold streams at any given point on the condenser's heat curve. The specific values of the approach and pinch temperatures were chosen to be equal to average values of these temperatures for commercial air condenser designs in Holt's database. Holt's database was used to determine air-condenser costs and parasitics.
- Condenser pressure drop was assumed to be 6 psi. This 6 psi pressure drop consists of 3 psi hydraulic drop and 3 psi to account for the presence of noncondensable gases in the working fluid loop. Holt's experience has shown that due to air leakage noncondensable gases build up in the working fluid loop effectively increasing the condensing pressure by a minimum of 3 psi.

Economic Analysis

The power plant capital cost was calculated by first calculating the cost of each major equipment item for the designed binary plant based on vendor data and proprietary Holt data and then using the major equipment costs along with the wellfield costs to calculate the total plant cost and the specific capital cost as discussed in Section 4.

A Holt model was used to design brine gathering and injection systems, and to estimate the gathering system installation and construction costs. Another Holt model was used to calculate parasitic power requirements for each submerged well pump. Details of these two Holt models are presented in Appendix A.

Results

Optimum geothermal power plants using the air-cooled commercial binary cycle were designed for Thermo Hot Springs, Raft River, Vale, Surprise Valley and Glass Mountain. Application of the air-cooled commercial binary cycle for the remaining sites was ruled out because at the higher temperature resources air-cooled binary cycles would not be as cost effective as dual flash cycles. Binary cycles include heat exchange equipent for transfering heat from the resource to the working fluid. This equipment adds a significant cost to the total plant cost and the additional heat transfer step adversely affects the cycle efficiency since it is non-isothermal and hence irreversible. Binary cycle plants were designed for Glass Mountain as a test case to compare binary plant costs with dual flash plant costs.

Specific power production curves for Thermo Hot Springs, Raft River, Vale, Surprise Valley and Glass Mountain are presented in Figure 5-2. Figure 5-2 indicates that cycle performance is a relatively flat function of turbine inlet pressure over the range of variables studied. The results used to plot Figure 5-2 were based on a plant output which was not adjusted to account for brine pumping, injection pumping and miscellaneous parasitics. It was felt that these parasitics would not affect the location of the economic optimum cycle. Thus, by not accounting for these parasitics optimum cycles could be determined quickly without affecting the results.

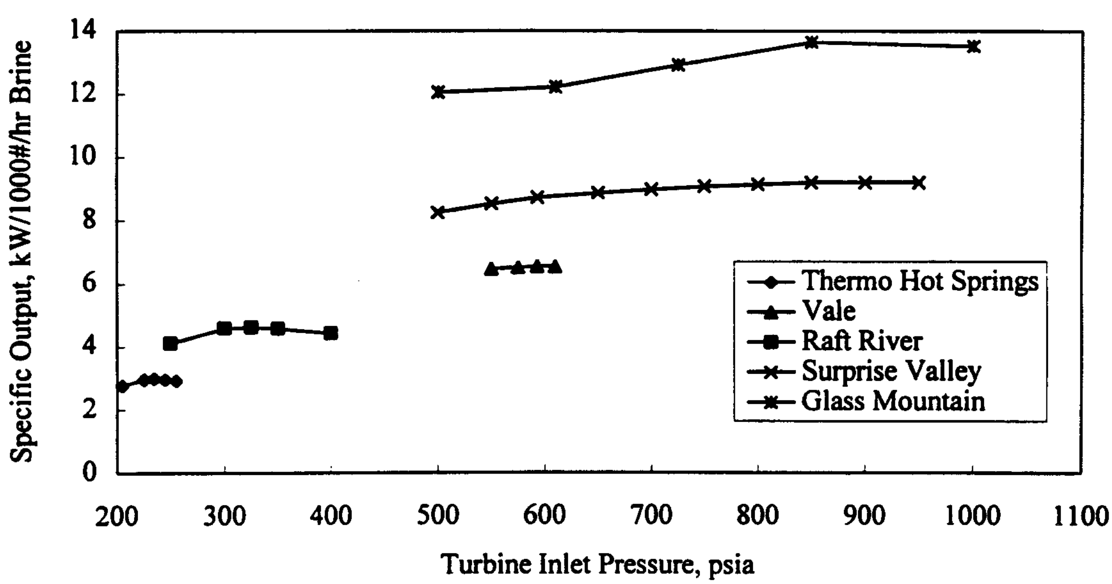

Figure 5-2
Air-cooled Commercial Binary: Specific Output Curves

For the air-cooled binary cycles it was assumed that the cycle which yields the maximum specific output is the optimum cycle. With this assumption optimum air-cooled binary cycles for individual sites were obtained using Figure 5-2. The specific output of the optimum cycles was then adjusted to account for brine pumping, injection pumping and miscellaneous parasitics. The adjusted specific output for the optimum cycles is plotted as a function of resource temperature in Figure 5-3.

Figure 5-3 shows that for the four sites with the coldest resources the specific output of air-cooled binary cycles increases almost linearly with temperature. This is an expected outcome since, apart from resource temperature, other important resource characteristics such as brine flow rate per well and minimum injection temperature are the same for all four sites.

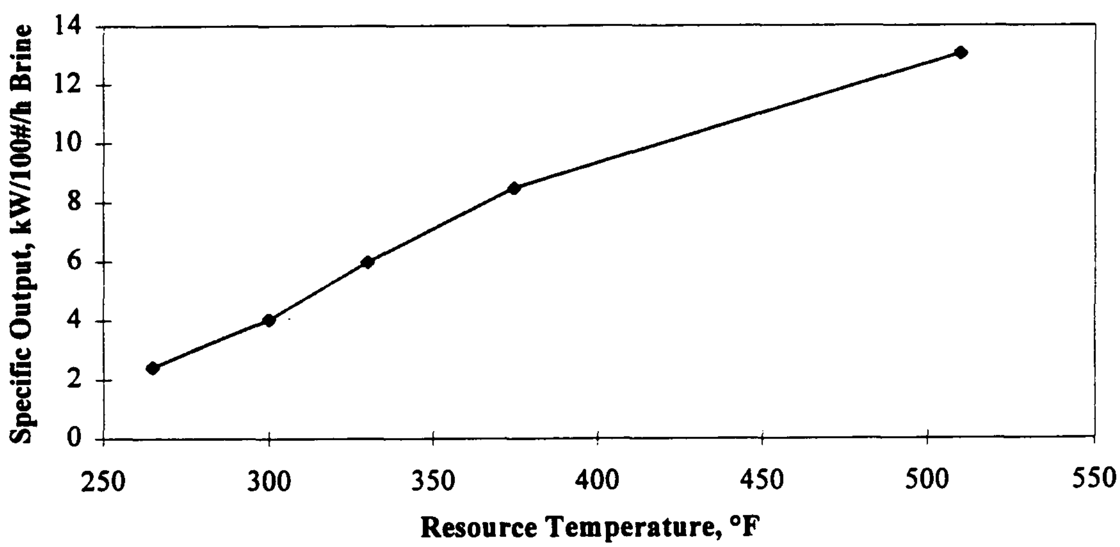

Figure 5-3
Air-cooled Commercial Binary Cycle: Specific Output v. Resource Temperature

Table 5-1 summarizes the results of the economic analysis of the air-cooled commercial binary cycle for various resources. Specific output (adjusted for brine production and injection pumping, and miscellaneous parasitics) and turbine inlet pressures for the optimum cycles are also listed in the table.

Total plant cost for air-cooled binary cycles is largely comprised of the costs for the wellfield, condenser (air-cooler), turbine, brine to working fluid heat exchanger, gathering system and pumps (production, injection and working fluid pumps). Table 5-1 lists the wellfield and condenser costs which add up to about 50% of the total plant cost. The total plant cost can be obtained from Table

5-1 by multiplying the specific cost by 50,000. Tables similar to Table 5-1 have been used to present cost results for other air-cooled binary cycles discussed in this report.

The specific capital cost for the air-cooled binary cycles is presented as a function of resource temperature in Figure 5-4. For the four coldest resources the specific capital cost decreases with increasing temperature which reflects the fact that cycle efficiency improves with increasing temperature, i.e. specific output increases as resource temperature increases. Consequently, wellfield cost and ultimately specific capital cost, is lowered as resource temperature increases because an increase in specific output decreases the total number of wells required for a 50 MW (net) plant.

Figure 5-4 shows that the specific capital costs of air-cooled binary cycles at Glass Mountain and Surprise Valley are almost equal although the Glass Mountain resource is significantly hotter than the Surprise Valley resource. The thermodynamic advantage of the hotter resource at Glass Mountain is largely lost for the following reason. The brine flow rate per well at Glass Mountain is significantly less than that at Surprise Valley (300,000 lb/h compared to 700,000 lb/h; Table 4-1) although the well costs are the same for the two sites. As a consequence, the wellfield cost for a 50 MW (net) plant at Glass Mountain is higher than that at Surprise Valley.

Table 5-1
Air-cooled Commercial Binary: Summary of Cases

Case	Wellfield Cost $1000	Condenser Cost $1000	Brine Rate 1000lb/h	Specific Output kWh/1000lb	Specific Cost $/kW
Vale, Oregon @ 330°F					
500 psia IC4 @85	33,441	31,726	8,678	5.76	2,440
610 psia IC4 @85	31,500	30,350	8,348	5.99	2,356
Surprise Valley, CA @ 375°F					
500 psia IC4 @84	31,000	30,616	6,800	7.35	2,166
850 psia IC4 @84	25,500	23,600	5,889	8.49	2,115
Thermo Hot Springs, UT @ 265°F					
205 psia IC4 @74	82,125	67,910	22,402	2.23	4,538
235 psia IC4 @74	76,875	60,499	20,482	2.44	4,188
Raft River, Idaho IC4 @85					
@325 psia	47,250	41,186	12,384	4.04	2,945
Glass Mountain, CA @ 510°F					
500 psia IC4 @87	38,500	29,395	4,257	11.75	2,142
800 psia IC4 @87	36,500	25,474	3,830	13.05	2,072

Note: Cases are identified as M psia IC4 @ N where M represents the turbine inlet pressure and N is the condensing temperature in °F.

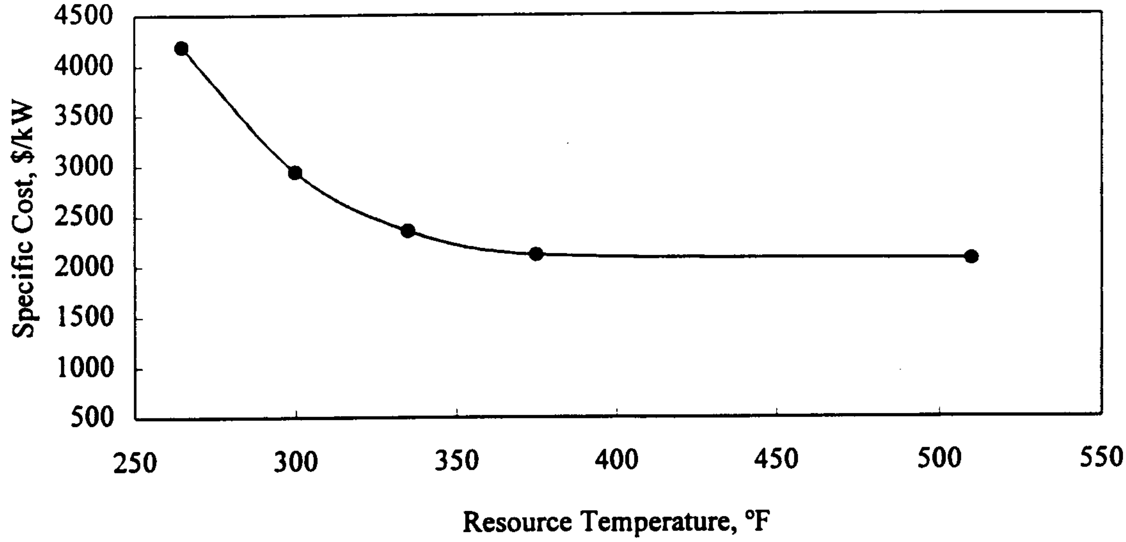

Figure 5-4
Air-cooled Commercial Binary Cycle: Specific Capital Cost v. Resource Temperature

The results of plant power production with the final resource temperature (end of run) are presented in Table 5-2. At reduced brine temperatures less working fluid is circulated so the auxiliary power consumption associated with pumping the working fluid is reduced. In spite of this reduction in hydrocarbon pumping parasitics, the net plant output at the final resource temperature is 20% to 50% lower than that at the initial resource temperature.

Water-cooled Commercial Binary

Introduction

The water-cooled binary cycle uses cooling water instead of air as the cooling medium. The water-cooled binary cycle is equivalent to the air-cooled binary cycle with the air condenser replaced by a water-cooled condenser plus a cooling tower and cooling water pumps. Water-cooled binary plants are attractive because they can produce more power than the air cooled binary plants during summer months. However, since water-cooled binary plants consume water, an ample supply of cooling water make-up is required. This study of water-cooled binary assumes that sufficient surface water is available at each resource.

Table 5-2
Air-cooled Commercial Binary Cycle: End of Reservoir Results

	Thermo Hot Springs	Raft River	Vale	Surprise Valley	Glass Mountain
Initial Temperature, °F	265	300	330	375	510
Final Temperature, °F	235	270	290	325	430
Initial Specific Output, kWh/1000 lb brine	2.44	4.04	5.99	8.50	13.05
Net at Final Temperature					
Generator Output, MW	46.753	45.371	36.727	43.318	52.443
Parasitics (inc. well pump), MW At Initial Temperature	24.704	18.548	17.677	17.237	14.136
Reduction in H/C Pump at final temperature, MW	2.33	2.452	4.755	3.163	1.827
Net Expected Output, MW	24.379	29.275	23.805	29.244	40.134
Power Loss at Final Temp., Percent	51.24	41.45	52.39	41.51	19.73
Final Specific Output, kWh/1000 lb brine	1.19	2.36	2.85	4.97	10.48

Cycle Process Flow

Figure 5-5 is a process flow diagram of the water-cooled commercial binary cycle. Since the basic process is similar to the air-cooled binary cycle the reader is referred to the air-cooled binary section of this report for the process flow description.

Performance Analysis

Power plants using the water-cooled binary cycle were evaluated for four sites: Surprise Valley, Thermo Hot Springs, Vale, and Raft River. Application of the water-cooled binary cycle to the remaining sites was ruled out because it was determined that dual flash cycles are more cost effective at hotter resources than binary cycles.

Description. Water-cooled binary cycles were modeled using the same methods as those used for modeling the air-cooled commercial binary cycles. The cooling tower size and parasitic loads were calculated using a Holt model that is described in Appendix A.

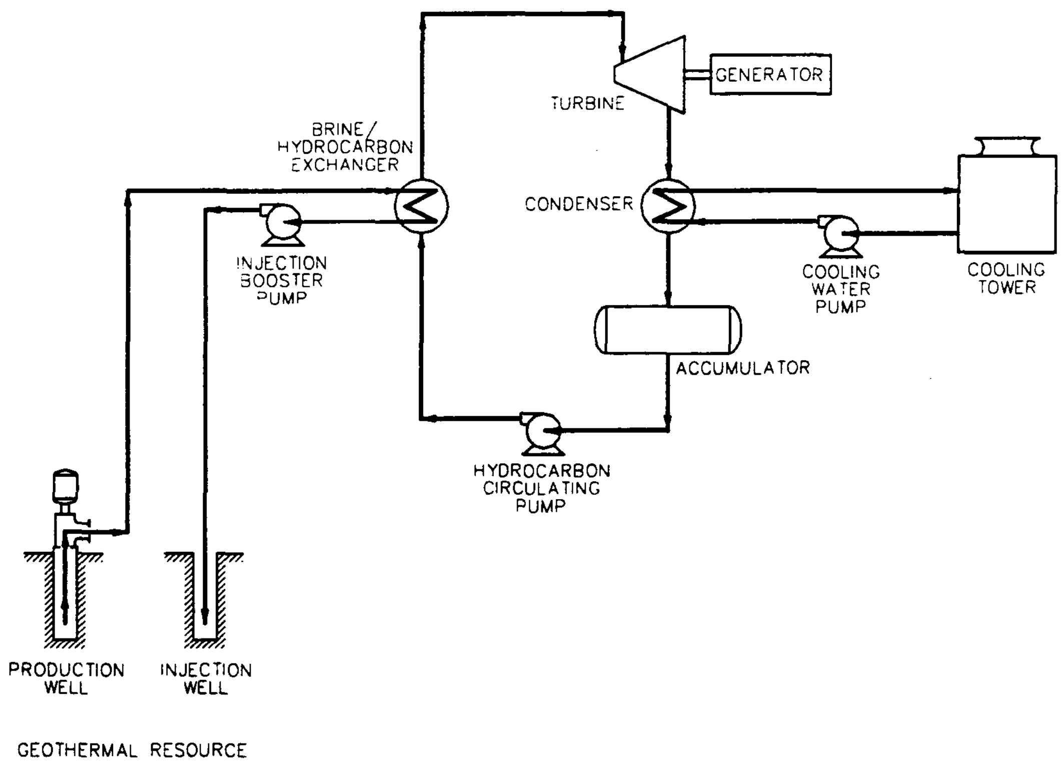

Figure 5-5
Water-cooled Commercial Binary Cycle: Process Flow Diagram

Assumptions. The assumptions used for modeling the water-cooled commercial binary cycle are similar to the assumptions used for modeling the air-cooled commercial binary cycles. Following are the specific assumptions used in the development of the water-cooled binary cycles:

- A temperature difference of 15°F between the working fluid and the cooling water was used for the condenser since it gave the minimum specific capital cost as shown in Figure 5-6.
- Cooling tower make-up water was assumed to be available at no cost because site-specific make-up water cost data was not available.
- The condensing temperature was taken as the average wet bulb temperature + 10°F cooling tower approach + 15°F cooling water rise + 15°F condenser hot end approach (i.e. wet bulb temperature + 40°F).
- The cooling tower is a counterflow type tower using high efficiency film fill.

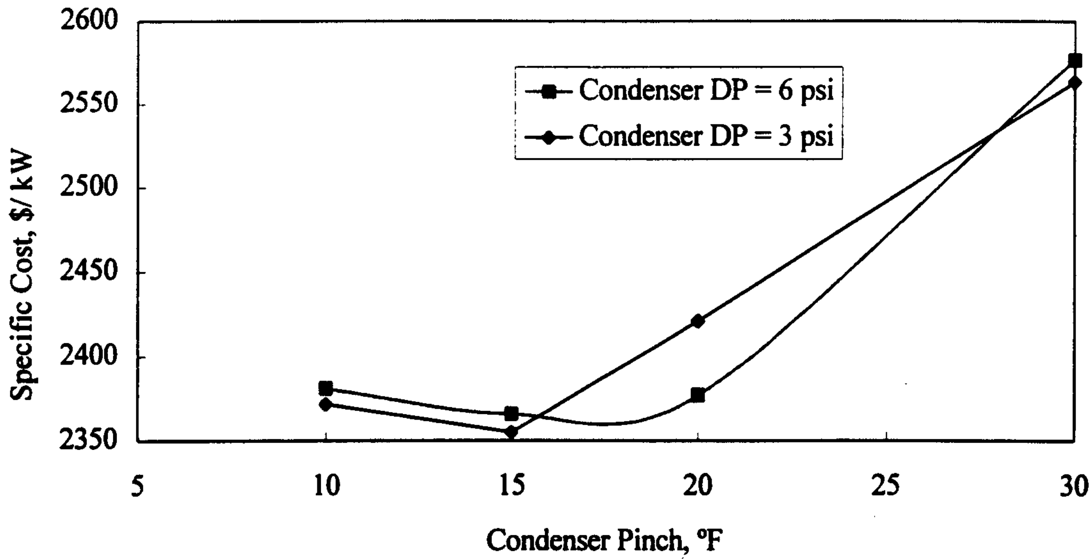

Figure 5-6
Specific Capital Cost v. Condenser Pinch, Vale

Cost Analysis

Water-cooled binary cycle power plants were evaluated by simulating six to ten cases for each site over a range of turbine inlet pressures. From these cases, four cases, corresponding to the highest specific output, were selected for equipment sizing and calculation of specific capital cost. The economic optimum was then found from a plot of specific capital cost versus operating pressure. For general details of cost analysis refer to the cost analysis section of the air-cooled binary cycle.

Results

Specific power production curves for Thermo Hot Springs, Raft River, Vale, and Surprise Valley are presented collectively in Figure 5-7. Figure 5-7 shows that the maximum specific output increases with increasing resource temperature. Also, the pressure at which the thermodynamic optimum occurs increases with increasing resource temperature. The corresponding specific capital cost curves are presented in Figure 5-8.

A comparison of Figures 5-7 and 5-8 indicates that the maxima in specific output correlate with the specific capital cost minima. Table 5-3 summarizes turbine inlet pressures for maximum specific output cases and minimum specific capital

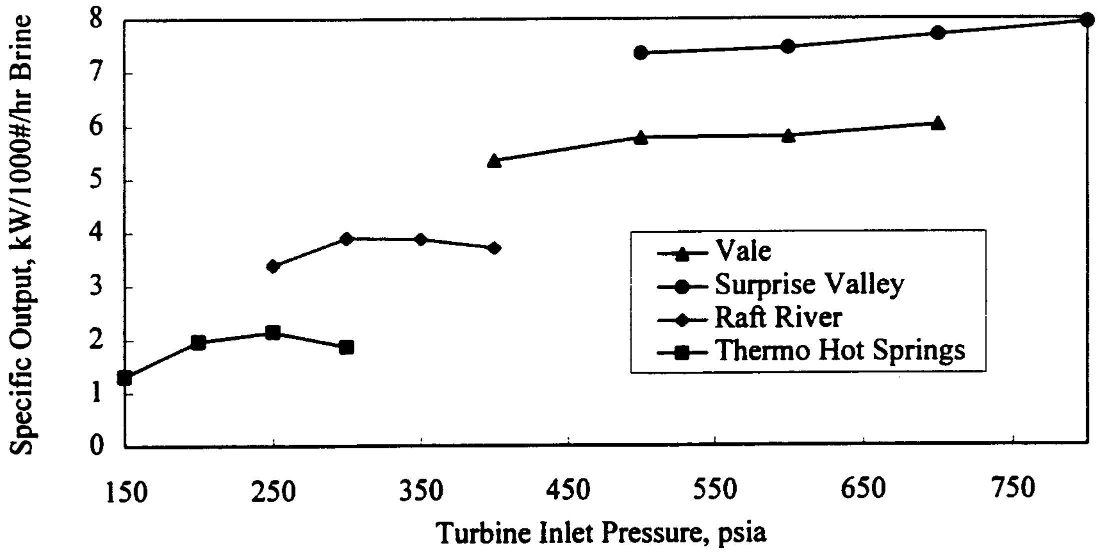

Figure 5-7
Water-cooled Binary: Specific Output v. Turbine Inlet Pressure

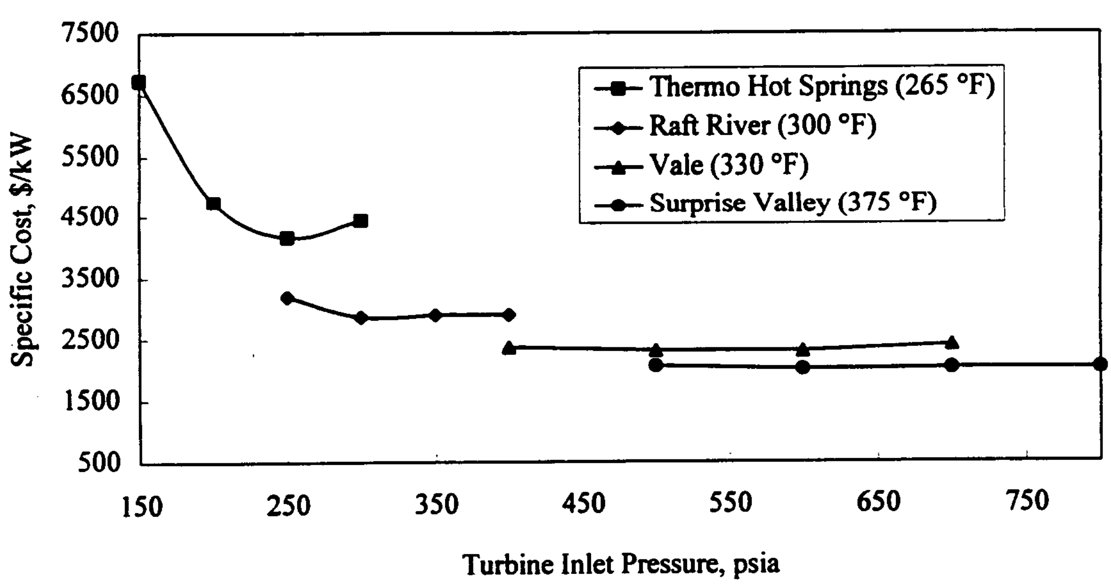

Figure 5-8
Water-cooled Binary: Specific Cost v. Turbine Inlet Pressure

Table 5-3
Turbine Inlet Conditions for Optimum Cases

Resource	Thermodynamic Optimum (psia)	Economic Optimum (psia)
Thermo Hot Springs	250	250
Raft River	300	300
Vale	700	600
Surprise Valley	900	600

cost cases. Turbine inlet pressures for the thermodynamic and economic optima coincide for Thermo Hot Springs and Raft River. For Vale and Surprise Valley the thermodynamic optima occur at relatively high turbine inlet pressures. As a result, at these two sites the maximum specific output cases do not yield minimum capital cost due to the fact that incremental increases in specific output are more than offset by increased costs associated with increases in pressures and decreases in exchanger LMTDs.

Table 5-4 presents a comparative summary of optimum cycles for the air-cooled and water-cooled commercial binary cycles. The table shows that the specific capital cost of water-cooled binary cycle power plants is only marginally less than that of air-cooled binary cycle plants. This result is somewhat unexpected because the heat removal system, the main distinction between the two cycles, for the water-cooled binary cycle is significantly less expensive than that for the air-cooled binary cycle. However, this cost advantage of the water-cooled binary cycle is largely offset by its higher parasitics compared to the air-cooled binary cycle. Water-cooled binary cycle parasitics for the cooling tower fans and cooling water pumps are higher than the air-cooled binary cycle parasitics for the fin fans. Consequently, the specific output of water-cooled binary cycles is less than that of air-cooled binary cycles. Also for these four resources, the annual average dry bulb temperature ranges from 37.9°F to 47.1°F. With dry, warmer weather conditions, water-cooled binary would have a larger cost advantage over air-cooled binary.

Commercial Dual Flash

Introduction

Dual flash technology finds wide application in the geothermal power industry. The analysis of dual flash technology serves to provide a useful baseline for evaluating competing NGGPP technologies. In fact, many of the next generation

Table 5-4
Comparison of Water-cooled and Air-cooled Binary Cycles

	Surprise Valley	Vale	Raft River	Thermo Hot Springs
Air-cooled Cases				
Number of Production Wells	9	12	18	29
Specific Output, kWh/1000lb brine	8.49	5.99	4.04	2.44
Turbine Inlet Pressure, psia	850	610	325	235
Specific Cost, $/kW	2115	2356	2945	4188
Water-cooled Cases				
Number of Production Wells	10	12	18	33
Specific Output, kWh/1000lb brine	7.45	5.80	3.90	2.15
Turbine Inlet Pressure, psia	600	600	300	250
Specific Cost, $/kW	2015	2302	2869	4164

technologies investigated in this study are modifications of the commercial dual flash process.

Cycle Process Flow

Figure 5-9 is a process flow diagram for the standard dual flash geothermal power plant. Geothermal brine flows through a throttle valve to a high pressure flash separator. High pressure steam from the separator overhead is expanded through the high pressure section(s) of one or more axial flow turbines which produce useful work. A portion of the high pressure stream from the separator is diverted to the vacuum system which uses steam ejectors to remove the noncondensable gases from the condenser.

Saturated liquid from the high pressure separator bottom flows through another pressure reducing valve to a low pressure separator. Low pressure steam from the separator overhead then flows to the turbine(s) where it is combined with partially spent high pressure steam to produce work in the low pressure turbine sections. The liquid from the low pressure separator bottom is pumped to the injection wells for return to the reservoir.

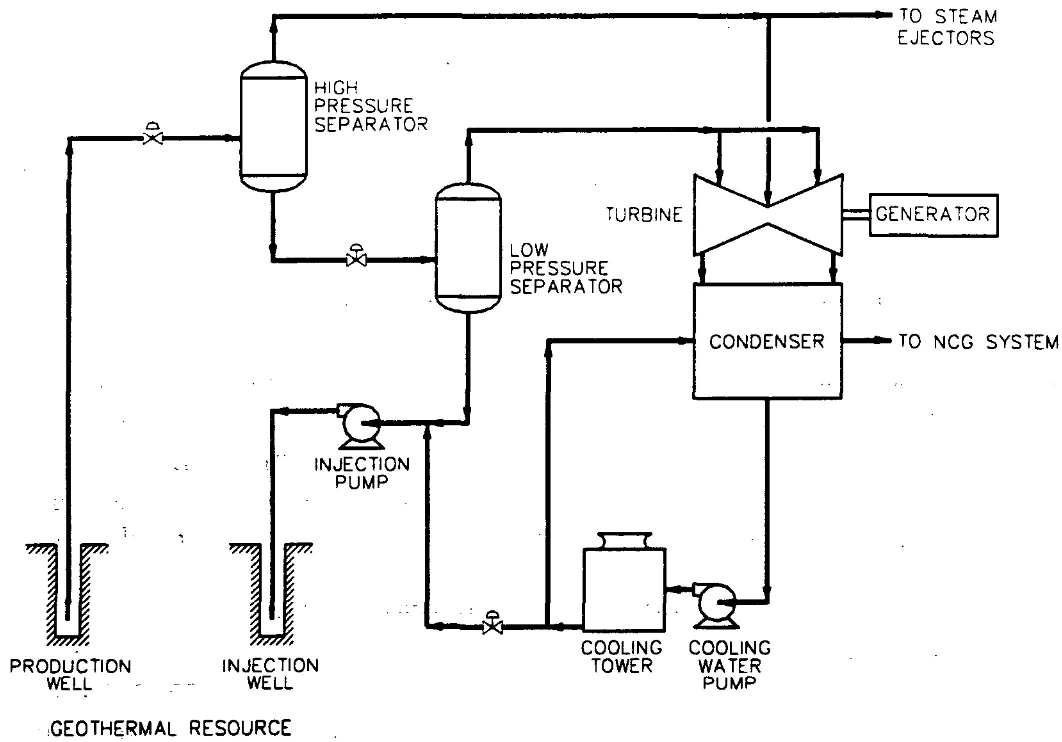

Figure 5-9
Commercial Dual Flash Cycle: Process Flow Diagram

Steam from the turbine exhaust is condensed in a water-cooled condenser. The condenser is usually a direct contact type condenser unless high H_2S flow mandates the use of a surface condenser. Most of the condensate and the cooling water returned from the condenser outlet is pumped back to the cooling tower. The excess liquid is injected into the geothermal reservoir along with the residual brine.

Noncondensable gases removed from the main condenser are discharged from the plant at roughly atmospheric pressure by the vacuum equipment. The gases are dispersed in the cooling tower fan stacks unless the H_2S concentration requires abatement. In this case, the gases are first sent to a liquid redox type sulfur plant. For the purposes of this study, the sulfur plant is assumed to be a liquid redox type plant. Liquid redox type sulfur plants are preferred in geothermal installations because the sulfur plant feed has a high CO_2 to H_2S ratio, and contains a relatively small amount of H_2S. Liquid redox processes have been claimed to be the most economical for removing small amounts of H_2S from large gas streams (Johnson et al, 1993; Kohl and Riesenfeld, 1985).

Performance Analysis

To assess its performance, dual flash technology was applied to all sites except Thermo Hot Springs and Clear Lake-Geysers. Both sites were eliminated by inspection because the resource at Thermo Hot Springs is a relatively cool resource (265°F), and Clear Lake is a hot dry rock resource.

Description. This study used an in-house Holt performance model to evaluate commercial dual flash technology. Geothermal power plants currently in operation have been designed using this model. Several dual flash power plant configurations were generated using the Holt model in order to locate the optimum dual flash power plant design for each geothermal site. Important model parameters that were optimized include the two flash pressures and condenser temperature.

Although flash pressures were varied to maximize specific output, the range of flash pressures was constrained. For free flowing resources the maximum flash pressure is set by the wellhead pressure. Wellhead pressures for these resources are listed in Table 4-4. Furthermore, the maximum high pressure flash pressure was limited to 153 psia since this study assumes that steam turbine inlet temperatures cannot exceed 360°F due to metallurgical limitations. Finally, low pressure turbine inlet pressures were constrained to a lower limit of one atmosphere to prevent air leakage into the process since this is the current industry standard practice.

Cooling water flow rate to the main condenser was varied to maximize power plant specific output. Increasing cooling water flow tends to increase power output by lowering condenser pressure. However, as condenser pressure falls the vacuum system load rises, increasing consumption of high pressure steam in the vacuum system. Also, the parasitic load of the cooling water pumps increases.

Assumptions. General modeling assumptions are listed in Table 4-1 and Table 4-3. The following data were obtained from these tables:

- Brine inlet and minimum rejection temperature
- Well costs
- Total dissolved solids content
- Site altitude
- Wet bulb and dry bulb temperatures
- Well flow rate for self-flowing wells.

The noncondensable gas content for each resource is listed in Table 4-5. In addition the following assumptions were made in modeling commercial dual flash power plants:

- Pressure drops in the steam lines from the high and low pressure separators to the turbine were set at 2.2 psi and 1 psi, respectively. In Holt's experience these pressure drops would correspond to optimum line sizes.
- The approach temperatures for the cooling tower and the condenser were constrained to be at least 5°F because it was felt that temperature differences less than 5°F could not be meaningfully measured.

The following values obtained from vendor data (Fuji Electric Co.) were used in this study:

- Stage group efficiencies (exclusive of mechanical and exhaust losses) of the high and low pressure turbine stage groups were taken to be 83% and 85%, respectively.
- The mechanical efficiency was assumed to be 97.5% to account for generator and bearing losses.
- Exhaust losses were calculated as a function of exhaust velocity. The largest allowable last stage blade diameter was 27 inches.
- Resulting overall efficiency varied from 76 to 78.5%.

Economic Analysis

Total plant costs for all technologies were calculated using the installation factor method: the installed cost is calculated by multiplying the major equipment cost by a site-specific plant cost factor. For dual flash power plants the value of the multiplier was 2.53 for all technologies. To determine the optimum dual flash plant configuration a number of cycle parameters were varied:

- Condenser type
- Vacuum system type
- Turbine number
- Cooling water circulation

The results of dual flash plant optimization are summarized in Tables 5-5 and 5-6. Important optimization parameters are discussed below.

Two types of condensers were considered for dual flash plants, surface condensers and direct contact condensers. The type of condenser to use in a dual flash plant depends on the hydrogen sulfide content of the geothermal fluid. Hydrogen sulfide contained in the turbine exhaust steam partitions into liquid and vapor phases in the condenser. The liquid phase hydrogen sulfide eventually migrates to the cooling tower where typically 50% of the dissolved H_2S is naturally abated (Gallup, 1994) and the rest is emitted. The 50% natural abatement in the cooling tower implies that H_2S emissions can be kept below the 11 lb/h limit if the cooling water leaving the condenser contains less than 22 lb/h of H_2S.

Table 5-5
Commercial Dual Flash Plant Process Description

Site	Temperature (F)	Production Well Type	Condenser Type	Vacuum System	No. of Turbines
Coso Hot Springs	525	Free	Surface	3 Stage	1
Desert Peak	425	Free	Direct	2 Jets	1
Dixie Valley	450	Free	Direct	2 Jets	1
Raft River	300	*Pumped*	Direct	3 Stage	2
Glass Mountain	510	Free	Direct	2 Jets	1
Salton Sea	570	Free	Direct	2 Jets	1
Surprise Valley	375	*Pumped*	Surface	2 Jets	2
Thermo Hot Springs	265	*Pumped*	Surface	3 Stage	2
Vale	330	*Pumped*	Surface	3 Stage	2

Table 5-6:
Commercial Dual Flash Plant Process Parameter Summary

Site	Brine Flow (M #/hr)	Turb in Press, 1 (psia)	Turb in Press, 2 (psia)	C.T. Circ. (gpm)	Cond. Aprch. (F)	Cond. Press. (" Hg)	No.of Prod. Wells	Gross Power (kW)	Net Power (kW)
Coso Hot Springs	3,200	130.0	24.0	50,000	7.5	3.03	8	56,359	51,995
Desert Peak	4,500	90.0	21.5	53,000	5.0	2.00	9	50,886	47,946
Dixie Valley	4,000	100.0	21.0	51,000	5.0	2.18	4	50,718	47,902
Raft River	15,392	27.0	12.5	100,000	5.0	1.79	22	60,656	47,023
Glass Mountain	3,000	151.0	26.0	53,000	5.0	2.08	10	54,174	51,339
Salton Sea	2,600	151.0	24.0	51,000	5.0	2.06	2	50,249	47,883
Surprise Valley	7,473	51.5	12.4	79,500	7.5	2.10	11	57,765	49,998
Thermo Hot Spr	37,903	20.0	11.9	124,000	5.0	2.43	53	78,325	49,921
Vale	10,375	35.4	12.7	96,000	7.5	1.72	15	58,603	48,159

The H_2S content of the condenser outlet cooling water stream mainly depends on geothermal steam composition and condenser type. For example, the presence of ammonia in geothermal steam increases H_2S partitioning in the liquid phase. (Weres, 1981). Direct contact condensers have much larger liquid to vapor ratios compared to surface condensers. Therefore, more turbine exhaust steam H_2S can dissolve in the liquid phase in a direct contact condenser than in a surface condenser.

Thus, prediction of the H_2S content of the cooling water stream requires a knowledge of the chemical composition of the geothermal steam at the various sites. In the absence of this information, this study assumed that 33% of the H_2S in the steam will partition into the liquid phase if a direct contact condenser is used to condense the turbine exhaust steam. As a result, direct contact condensers were used for those resources which would yield H_2S flow rates less than 66 lb/h in the steam flow required to produce 50 MW (net) of power.

The cost of direct contact condensers is a function of the total duty whereas surface condenser costs are a function of surface area. The condenser approach temperatures were also optimized for each resource since reducing the approach temperature increases power output but also increases the required condenser surface area. As an illustration, the specific plant cost versus condenser approach temperature is plotted for Vale, Oregon in Figure 5-10. The figure

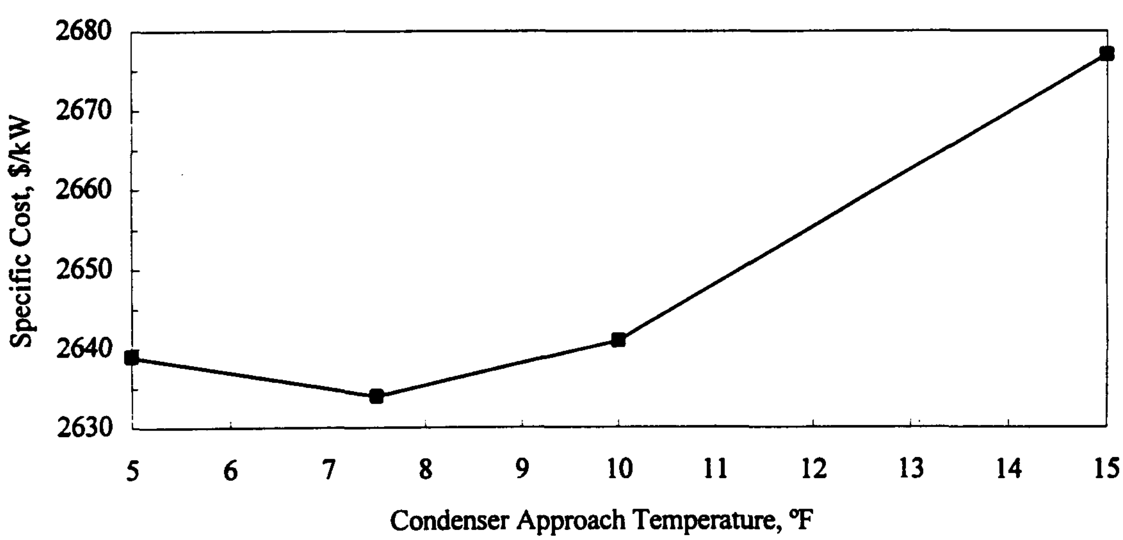

Figure 5-10
Condenser Approach Optimization, Vale

shows that 7.5°F is the most economical condenser approach temperature for Vale. The optimum approach temperatures for each site are listed in Table 5-6. Vacuum system type is another optimization parameter used in the design of dual flash plants. Two vacuum system configurations were considered:

- two stage steam ejector system
- three stage hybrid system, with a vacuum pump as the third stage

In general, higher specific output is obtained with the three stage hybrid system compared to the two stage ejector system but the vacuum pump adds to the cost. The three stage hybrid system is cost effective when the vapor/gas flow rate into the condenser is high for low temperature resources and for resources with high noncondensable gas concentration.

Figure 5-11 compares the specific plant cost for plants using the two types of vacuum system for several resources. It can be seen that the two systems are equally cost effective at a resource temperature of 375°F (Surprise Valley). For resource temperatures below 375°F, plants using the three stage hybrid system are somewhat lower in cost than plants using two stage jets; the converse is true for resource temperatures above 375°F.

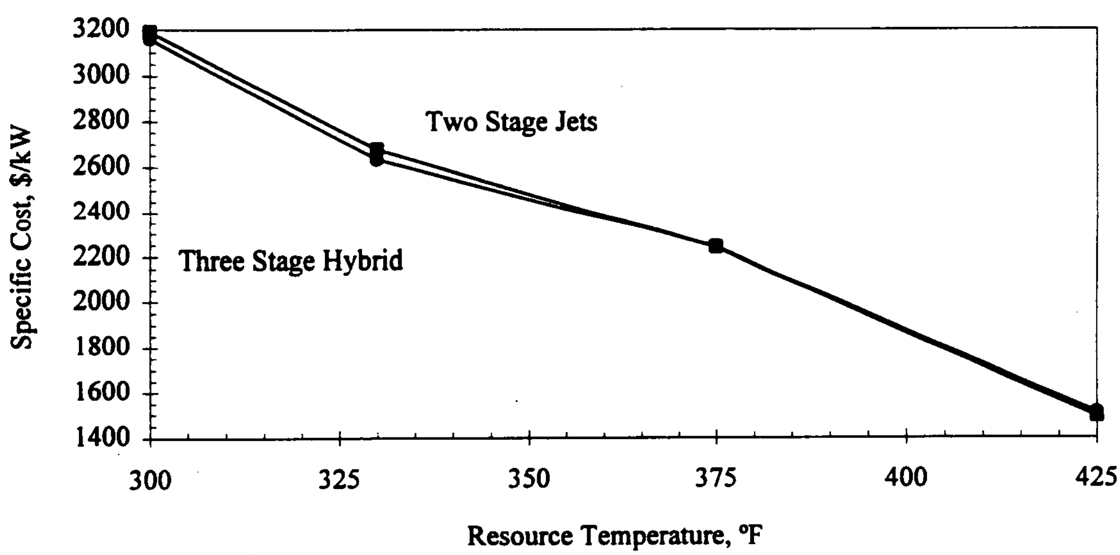

Figure 5-11
Comparison of Vacuum Systems

Cooling water circulation rate was adjusted to optimize condenser operation. A plot of cooling water circulation versus specific plant cost is shown on Figure 5-12 for Vale, Oregon. The optimum cooling water flow rate is 96,000 gpm. Similarly, optimum cooling water flow rates were determined for each site and are listed in Table 5-6.

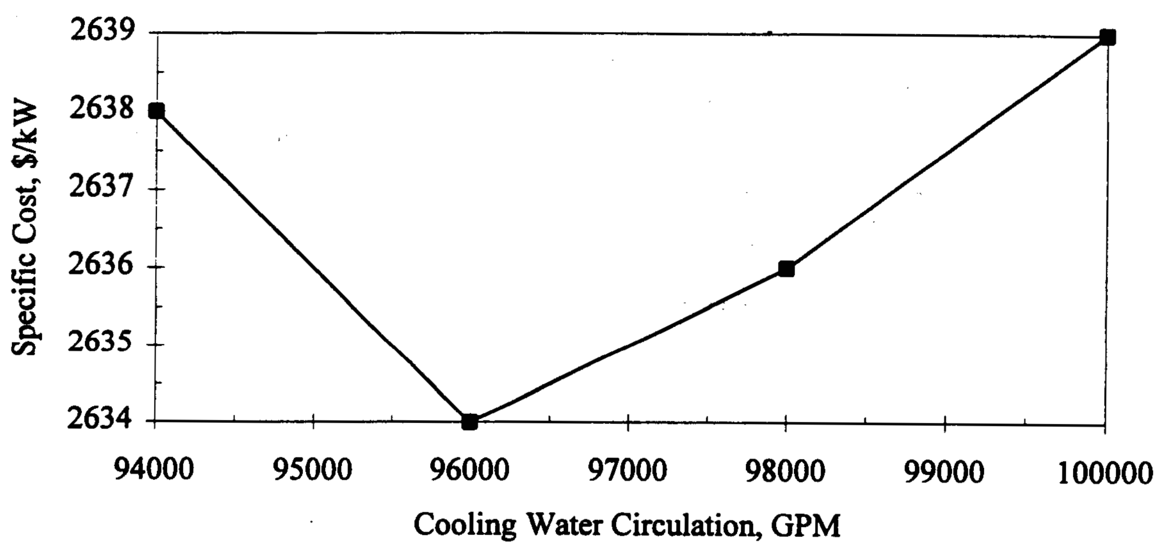

Figure 5-12
Cooling Water Circulation Optimization, Vale

Results

Performance. Specific output for various resources is shown on Figure 5-13. It can be seen from the figure that a dual flash plant at Salton Sea (570°F) would yield the maximum specific power output (18 kW/1000 Lb/hr); the minimum specific power output (1.2 kW/1000 lb/hr) would be obtained at Thermo Hot Springs (265°F).

Figure 5-13 shows that, in general, specific output increases with resource temperature. However, the characteristics of individual resources have a significant effect on specific output. For example, the geothermal brine at Coso Hot Springs contains a larger amount of noncondensable gases (2 % wt) which affects the size and cost of the vacuum system. Consequently, the specific output of a dual flash plant at Coso is lower than that for a resource of equal temperature but lower non-condensable gas content. Similarly, the presence of

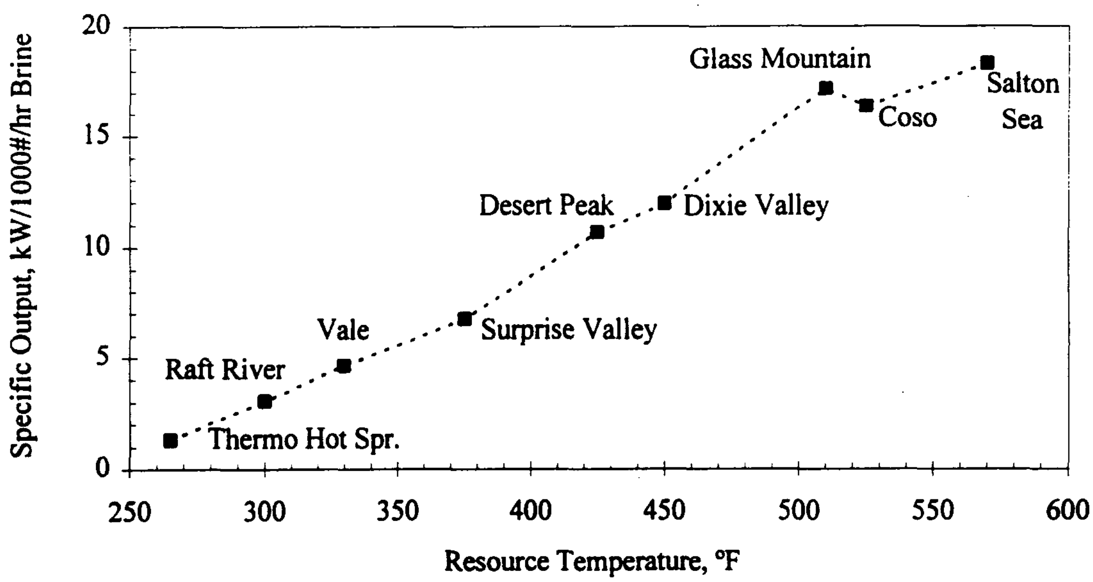

Figure 5-13
Commercial Dual Flash Cycle: Economic Optimum Specific Output v. Resource Temperature

large amounts of dissolved solids (15-25 % wt) in the brine has an adverse effect on the specific output of a dual flash plant at Salton Sea.

Resources below 400°F are pumped resources. The well pumping power requirement reduces the specific output for these resources relative to resources with free flowing wells. Moreover, the low pressure flash pressure for these cooler resources is constrained to a lower limit of one atmosphere which may not be thermodynamically favorable.

Economics. Using the optimum configurations listed in Table 5-5 and Table 5-6, specific capital costs (for a nominal 50 MW plant) were obtained for each resource and are presented in Figure 5-14. Some general observations on the dependence of specific capital cost on resource types are discussed below.

In general, specific capital cost is dependent on resource temperature and other characteristics in a manner similar to specific output as discussed above. Thus, for example, although Coso Hot Springs is a relatively hot resource, a dual flash plant at Coso would have a higher specific capital cost due to the presence of larger amounts of non-condensable gases and H_2S in the brine.

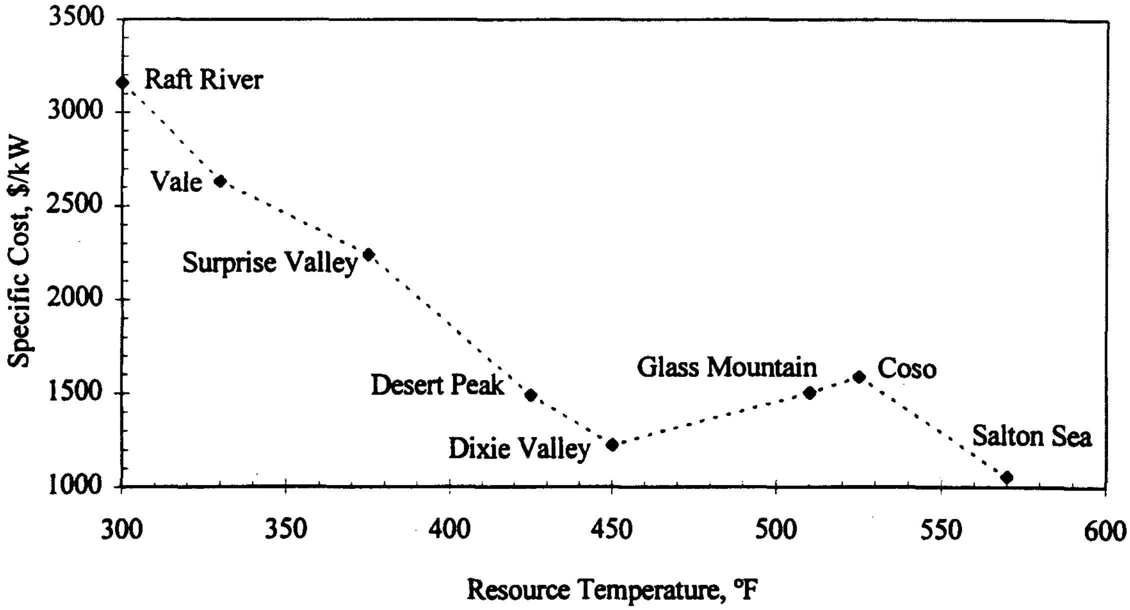

Figure 5-14
Commercial Dual Flash Cycle: Specific Capital Cost v. Resource Temperature

For self-flowing wells, the average well flow rate has a significant effect on capital cost since well flow rate largely determines the number of wells, and low well flow rates translate into larger wellfield costs. This phenomenon is highlighted by comparing the specific capital costs for Dixie Valley (450°F resource) and Glass Mountain (510°F resource). The well flow rate at Dixie Valley is more than three times that at Glass Mountain. As a consequence, even though Glass Mountain is a significantly hotter resource than Dixie Valley, it would have a higher plant cost.

Finally, resources below 400°F are pumped and the loss of power due to pumping parasitics increases specific plant cost. Thus, the specific plant cost for the cooler resources (<400°F) is significantly higher than that for the hotter resources (>400°F).

Equipment and total plant costs for an optimal dual flash power plant at various sites are presented in Table 5-7 which presents an overview of the important capital cost elements of a dual flash plant and their interrelationships. For example, it shows that the cost of the turbine generator set comprises about 60% of the total installed equipment cost. The turbine generator cost for a plant at Coso is only about 52% of the total installed equipment cost because a plant at

Table 5-7
Commercial Dual Flash Cycle: Major Equipment and Total Plant Costs

CASE	Coso Hot Sp. 525°F	Desert Peak 425°F	Dixie Valley 450°F	Raft River 300°F
Power Plant				
H.P. Separators	107,802	143,397	135,600	638,400
L.P. Separators	140,000	165,000	165,000	412,500
Purifiers	39,700	36,600	38,700	129,600
Silencers	22,500	22,500	22,500	22,500
Condenser	2,676,300	1,625,600	1,607,000	2,157,100
Hot Well/Cond. Pumps	95,600	880,400	856,700	1,361,000
Fire Pumps	55,000	55,000	55,000	55,000
L. O. Trans. Pumps	4,000	4,000	4,000	4,000
Pot. Water Pumps	7,000	7,000	7,000	7,000
Aux. C.W. Pumps	542,100	29,700	44,500	54,800
Injection Pumps	75,700	103,200	93,900	334,700
Cooling Tower	1,193,600	1,265,200	1,217,500	2,387,200
Plant Air System	129,300	119,200	119,100	116,900
L. O. Storage Tanks	29,000	29,000	29,000	29,000
Turbine Generator	12,120,000	11,758,000	11,700,000	19,723,000
NCG Removal	1,605,400	75,000	86,400	3,039,700
Gantry Crane	500,000	500,000	500,000	500,000
Vac Hot Well	15,100	15,100	15,100	15,100
Sulfur Plant	2,965,000	488,000	549,000	0
Misc. Tanks	184,000	157,000	156,000	285,000
Start-up or Emer Gen.	31,000	31,000	31,000	31,000
FW Tank + Sys	160,263	169,875	163,463	320,513
Total Major Equipment	23,302,226	18,247,748	18,161,090	32,623,171
Installed Plant Cost	58,955,000	46,167,000	45,948,000	82,537,000
Gathering & Injection System				
Production Pumps	0	0	0	3,123,295
Prod. Pump Aux.	0	0	0	158,400
Silencers	25,000	25,000	25,000	25,000
Total Major Equipment	25,000	25,000	25,000	3,306,695
Total	53,000	53,000	53,000	6,977,000

CASE	Coso Hot Sp.	Desert Peak	Dixie Valley	Raft River
Summary				
Plant Equip. Cost	23,302,226	18,247,748	18,161,090	32,623,171
Installed Plant Cost	58,955,000	46,167,000	45,948,000	82,537,000
Gath & Injec Equip Cost	25,000	25,000	25,000	3,306,695
Installed Gath & Injec Cost	53,000	53,000	53,000	6,977,000
Gath & Injec Piping Cost	1,540,000	1,486,000	1,112,000	1,402,000
Well Cost	22,000,000	23,765,000	11,571,000	57,750,000
Total Plant Cost	82,548,000	71,471,060	58,684,429	148,666,000
Specific Cost ($/kW)	1,588	1,491	1,225	3,161

Table 5-7 (Continued)
Commercial Dual Flash Cycle: Major Equipment and Total Plant Costs

CASE	Glass Mtn 510°F	Salton Sea 570°F	Surprise Val 375°F	Vale, Oregon 330°F
Power Plant				
H.P. Separators	101,248	67,800	319,200	399,000
L.P. Separators	140,000	82,500	412,500	412,500
Purifiers	43,000	19,300	92,200	101,500
Silencers	22,500	22,500	22,500	22,500
Condenser	1,542,700	1,429,200	3,244,500	3,290,500
Hot Well/Cond. Pumps	880,300	857,800	112,600	113,200
Fire Pumps	55,000	55,000	55,000	55,000
L. O. Trans. Pumps	4,000	4,000	4,000	4,000
Pot. Water Pumps	7,000	7,000	7,000	7,000
Aux. C.W. Pumps	42,100	40,500	716,000	801,700
Injection Pumps	75,300	66,700	146,500	214,300
Cooling Tower	1,265,200	1,217,500	1,897,800	2,291,700
Plant Air System	127,600	118,900	124,300	119,700
L. O. Storage Tanks	29,000	29,000	29,000	29,000
Turbine Generator	11,860,000	11,492,000	18,903,000	19,260,000
NCG Removal	75,000	75,000	198,300	1,058,700
Gantry Crane	500,000	500,000	500,000	500,000
Vac Hot Well	15,100	15,100	15,100	15,100
Sulfur Plant	1,219,000	977,000	1,986,000	2,341,000
Misc. Tanks	157,000	151,000	251,000	265,000
Start-up or Emer Gen.	31,000	31,000	31,000	31,000
FW Tank + Sys	169,875	163,463	254,813	307,688
Total Major Equipment	18,930,373	17,966,356	30,232,982	32,582,992
Installed Plant Cost	47,894,000	45,455,000	76,489,000	82,435,000
Gathering & Injection System				
Production Pumps	0	0	1,559,149	2,126,113
Prod. Pump Aux.	0	0	79,200	108,000
Silencers	25,000	25,000	25,000	25,000
Total Major Equipment	25,000	25,000	1,663,349	2,259,113
Total	53,000	53,000	3,510,000	4,767,000

CASE	Glass Mtn	Salton Sea	Surprise Val	Vale, Oregon
Summary				
Plant Equip. Cost	18,930,373	17,966,356	30,232,982	32,582,992
Installed Plant Cost	47,894,000	45,455,000	76,489,000	82,435,000
Gath & Injec Equip Cost	25,000	25,000	1,663,349	2,259,113
Installed Gath & Equip Cost	53,000	53,000	3,510,000	4,767,000
Gath & Injec Piping Cost	1,754,000	542,000	1,084,000	1,194,000
Well Cost	27,500,000	4,500,000	31,000,000	38,471,000
Total Plant Cost	77,201,000	50,550,000	112,083,000	126,866,588
Specific Cost ($/kW)	1,504	1,057	2,242	2,634

Coso requires a sulfur plant and a relatively expensive condenser. In a similar vein, due to high well flow rates at Salton Sea, the wellfield cost is less than 10 % of the total capital cost whereas at other resources wellfield cost is 20 to 40 % of the total capital cost.

Conclusions

In conclusion, it would be instructive to compare the performance of the three baseline technologies at the various geothermal sites. Figures 5-15 and 5-16 are plots of economic optimum specific output and specific capital cost, respectively, for the air-cooled commercial binary, water-cooled commercial binary, and dual flash cycles for all the geothermal sites that the cycles were developed for. Figure 5-16 shows that for resources hotter than Surprise Valley, dual flash power plants have a lower specific capital cost than binary power plants. For Surprise Valley and colder resources, binary power plants have lower specific capital costs compared to dual flash power plants. It appears, then, that binary cycle is generally the current technology of choice for resources colder than about 400°F; whereas for resources hotter than 400°F dual flash technology is favored. This is consistent with the development history of geothermal power to date.

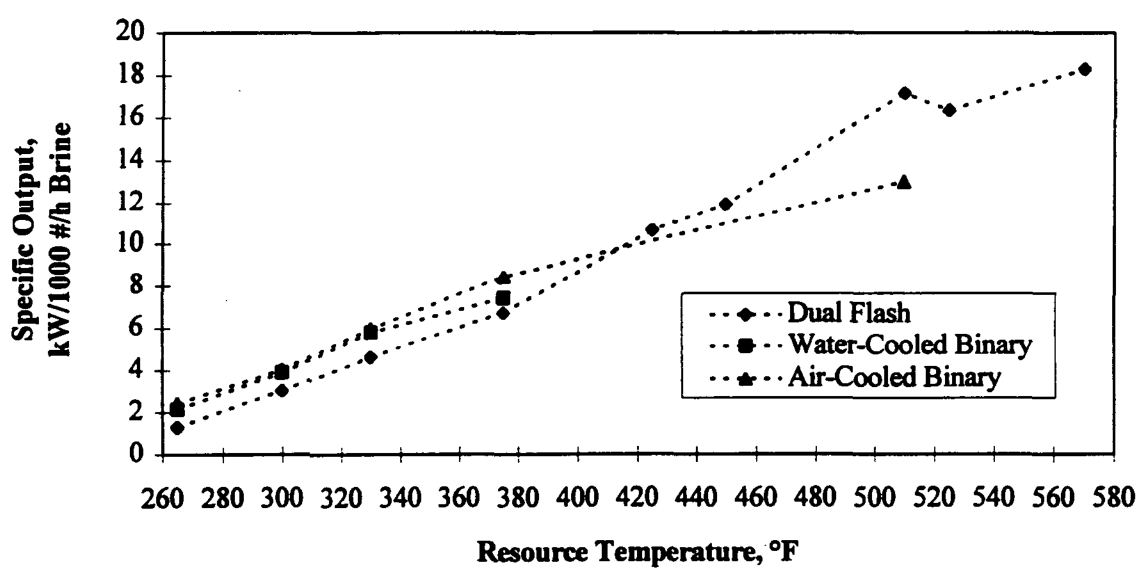

Figure 5-15
Baseline Technologies - Specific Output v. Resource Temperature

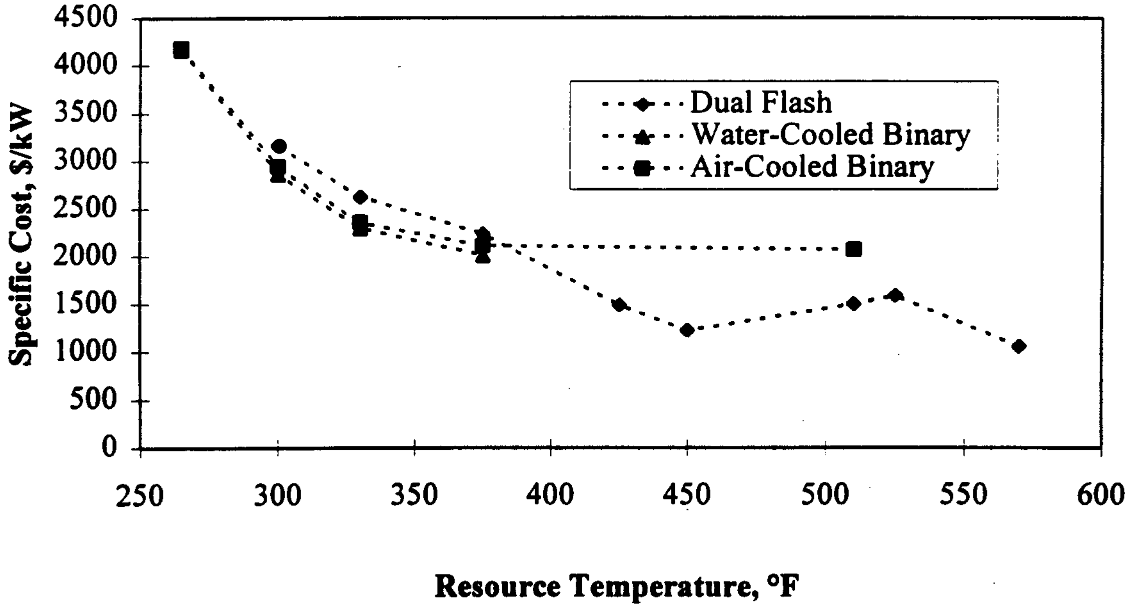

Figure 5-16
Baseline Technologies - Specific Capital Cost v. Resource Temperature

6

ADVANCED BINARY CYCLES

Mixed Fluid Binary: Hydrocarbon Mixtures

Introduction

As noted earlier, both air-cooled and water-cooled commercial binary cycles were evaluated with commercial isobutane as the working fluid. Prior studies have suggested that using mixtures of hydrocarbons instead of pure components in binary cycles will enhance the cycle's specific output (Demuth, 1982; Bliem and Mines, 1993). In this section of the report the evaluation of binary cycles with mixed working fluids is presented.

The selection of working fluid mixtures was primarily based on literature and Holt's own experience. In conformity with other work, only hydrocarbon mixtures with isobutane as the predominant component were used. For example, for a 360°F resource, a 96% isobutane/4% heptane mixture has been reported to be the most promising of three hydrocarbon mixtures (Demuth and Kochan, 1982).

This study used a 94 wt % isobutane/6 wt % heptane (94/6) mixture as its focus. A more dilute isobutane concentration than that reported by Demuth and Kochan was used because using a 96 wt % isobutane mixture would have made the mixture nearly identical to commercial isobutane which is 96.5 wt % isobutane. Several mixtures other than the 94/6 mixture were evaluated for the various sites. For the hotter resources the isobutane-heptane mixtures were diluted with hydrocarbons which have a higher molecular weight than isobutane. Conversely for Raft River and Thermo Hot Springs, isobutane-heptane mixtures diluted with propane were evaluated.

In binary cycles that use mixed working fluids, a non-isothermal condensing curve is obtained which enables the use of smaller air-cooled condensers compared to commercial binary cycles. As a result, the economic optimum mixed fluid cycles generally yield a lower specific capital cost than commercial binary cycles.

It is worth noting here that past studies which used mixtures of hydrocarbons as working fluids primarily focused on counterflow water-cooled binary units that assumed a minimum temperature difference of 10°F (Bliem and Mines, 1993). For some mixed fluid cycles with non-isothermal condensing curves, a temperature difference of 10°F between the fluids on the hot end of the exchanger was used. In this study, mixed fluids were evaluated using crossflow air-cooled cycles in which condenser approaches of 10°F are not generally achievable.

Cycle Process Flow

The mixed fluid binary cycle analyzed in this study has the same configuration as the air-cooled commercial binary cycle. Hence, the reader is referred to Section 5 for the process flow description and diagram (Figure 5-1).

Performance Analysis

Description. Mixed fluid cycles were evaluated using essentially the same methodology as that used to evaluate the air cooled commercial binary cycles. Hence, once again, the reader is referred to Section 5.

Assumptions. The specific and general assumptions used in the analysis of the air-cooled commercial binary cycle also apply to the mixed fluids binary cycle (refer to Section 5).

Heat transfer coefficients used to calculate the sizes and costs of the heat exchangers for the air-cooled commercial binary cycle were also used for heat exchanger sizing for the mixed fluid binary cycle. Since there is some concern that heat transfer coefficients of mixed fluids are not as high as for the "pure" fluids used in the commercial binary cycles, a sensitivity analysis was performed to determine the effect of a 10% reduction in heat transfer coefficients on the specific power cost. The results of the sensitivity analysis are summarized in Table 6-1. It can be seen that a 10% reduction in heat transfer coefficients affects the plant cost by 1 to 3%. Thus, it appears that the assumption is justified and a 10% variation in heat transfer coefficients only affects the plant cost marginally.

Table 6-1
Effect of Varying Heat Transfer Coefficient on Specific Plant Cost

No.	Case	% Difference	Specific Cost, Best Cycle ($/kW)	Specific Cost, 90% U ($/kW)
1	Surprise Valley	2.80	1859	1911
2	Vale	3.07	2184	2251
3	Raft River	1.03	2701	2774
4	Thermo Hot Springs	2.86	3919	4031

Cost Analysis

The cost model used for calculating plant costs for the air-cooled commercial binary cases was also used for calculating plant costs for the mixed fluids binary cycles.

Results

Table 6-2 compares the specific capital cost and specific output of the economic optimum mixed fluid cycle with those of the optimum air-cooled commercial binary cycle. Mixed fluid cycles have a lower specific capital cost than their commercial binary counterparts even though they have a lower specific output. The lower specific output for mixed fluid cycles is offset by the lower condenser costs compared to the commercial binary cycles. Table 6-3 provides a summary of major equipment and total plant costs for the optimum mixed fluids cycles. Detailed results of the evaluation of the mixed fluid cycle performance for individual sites are presented below.

<u>Surprise Valley, California</u>
Past studies (Bliem and Mines, 1993) have shown that 350-400°F resource temperatures are well suited to mixed fluid cycles, and Surprise Valley with a resource temperature of 375°F falls in that range. Mixed fluids cycles at Surprise Valley were evaluated for the following working fluid mixtures:

- 80 mole % iC4/20 mole % iC5 (80/20 mixture)
- 94 wt % iC4/6 wt % iC7 (94/6 mixture)

Table 6-4 summarizes the performance of the mixed fluid cycles at Surprise Valley. It can be concluded from the table that the 94/6 mixture yields the minimum capital cost whereas the commercial iC4 mixture yields the maximum specific output. The same conclusion can be drawn from Figure 6-1 and Figure 6-2 which are plots of specific output and specific capital cost, respectively.

Table 6-2
Comparison of Mixed Fluid Cycle with Air-cooled Commercial Binary Cycle

Resource	Specific Output (kW/1000 lb/hr brine)		Specific Capital Cost ($/kW)	
	Commercial Binary	Mixed Fluid	Commercial Binary	Mixed Fluid
Vale, Oregon	5.99	5.26	2356	2184
Surprise Valley	8.49	7.55	2115	1859
Raft River	4.04	3.52	2945	2701
Thermo Hot Springs	2.44	2.18	4188	3919

Table 6-3
Mixed Fluids Equipment and Total Plant Cost Summary

	Thermo Hot Springs	Raft River	Vale	Surprise Valley
Accumulator	351,283	285,129	258,751	231,851
Brine/HC Preheater	2,190,155	2,438,370	3,930,875	2,591,049
Brine/HC Vaporizer	2,327,309	1,015,146	777,293	1,245,670
Air Condenser	11,609,932	7,734,853	6,082,105	5,249,391
Turbine Generator Set	12,790,881	11,786,609	11,558,847	11,809,938
I-C4 Pump	794,720	833,873	1,019,689	1,352,814
Injection Pumps	278,219	183,531	122,804	93,774
Well Pumps	5,191,863	3,113,084	1,918,043	1,351,387
Total Equipment Cost	35,534,362	27,390,596	25,668,409	23,925,874
Multiplier	3.10	2.96	2.83	2.80
Installed Plant Cost	110,156,525	81,076,165	72,641,597	66,992,446
Production Well Drilling	60,000,000	37,500,000	24,705,882	20,000,000
Injection Well Drilling	24,000,000	15,000,000	10,500,000	7,500,000
Gathering System	1,817,000	1,457,000	1,333,187	1,104,000
Net Plant Output (MW)	50	50	50	50

Table 6-4
Surprise Valley Mixed Fluids Case Summary

Case	Wellfield Cost $1000	Condenser Cost $100	Brine Flow 1000 lb/hr	Specific Output kWh/ 1000 lb brine	Specific Capital Cost $/kW
<u>80/20 iC4:</u>					
500 psia	26,250	23,224	6,661	7.51	1,918
600 psia	26,250	22,398	6,461	7.74	1,932
700 psia	26,250	21,721	6,291	7.95	1,950
850 psia	24,375	20,664	6,038	8.28	2,007
<u>94 iC4/6 C7:</u>					
500 psia	31,000	16,338	7,056	7.09	1,895
600 psia	31,000	15,995	6,914	7.23	1,839
700 psia	27,500	15,242	6,620	7.55	1,859
850 psia	27,500	14,698	6,358	7.86	1,912

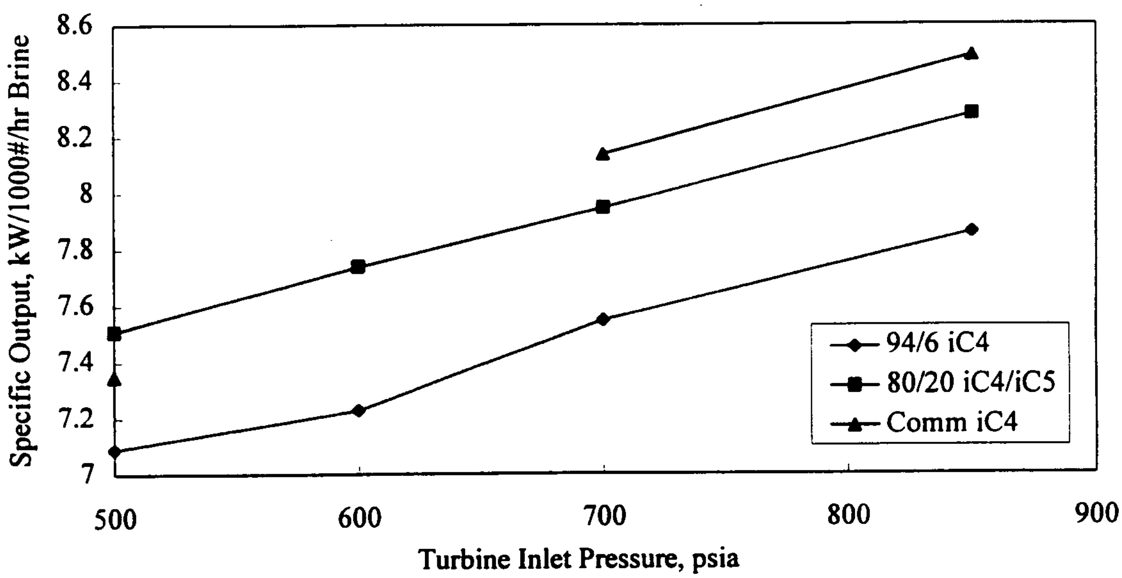

Figure 6-1
Mixed Fluids Specific Output Curves, Surprise Valley

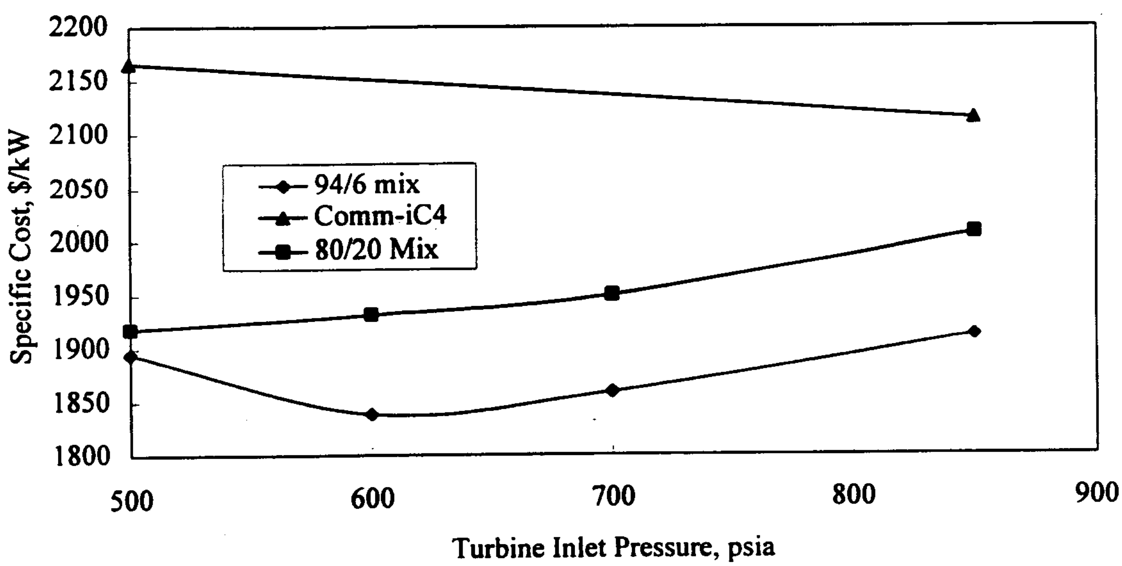

Figure 6-2
Mixed Fluids Specific Capital Cost Curves, Surprise Valley

In the course of evaluating mixed fluids cycles at Surprise Valley, the issue of optimum condenser approach and pinch was investigated. A number of mixed fluids cycles were simulated with the 94/6 mixture at 600 psia turbine inlet pressure and with varying cold end approaches to determine the optimum cold end pinch. The results of this optimization are plotted in Figure 6-3 which shows that a cold end approach of about 30°F is optimum for the 94/6 mixture. Thus, the 34°F cold end approach used in this study, so as to better compare results with the commercial isobutane cases, is optimal or near optimal.

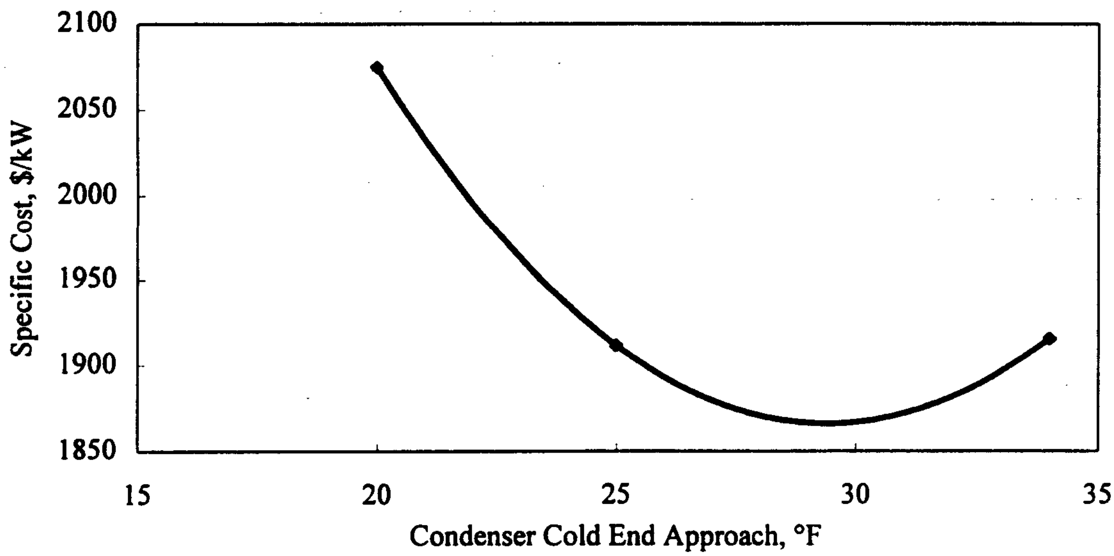

Figure 6-3
Specific Capital Cost v. Cold End Approach, Surprise Valley

As noted earlier, a 10°F cold end approach has been used in previous studies on counterflow water-cooled mixed fluid binary cycles (Demuth, 1982; Bliem and Mines, 1993). For the Surprise Valley resource, a 10°F cold end approach leads to a temperature cross in the air-cooled condenser, and a feasible air cooler design cannot be obtained. Since a water-cooled condenser operating in pure counterflow will have a significantly different thermal performance than an air-cooled condenser which is a crossflow exchanger, there will be a difference between our results and those of Bliem and Mines.

Vale, Oregon

Mixed fluid cycles at Vale were only evaluated for the 94/6 mixture, and Table 6-5 summarizes cycle performance at Vale for the 94/6 mixture. Using the data in Table 6-5, specific output and specific capital cost are plotted in Figure 6-4 and Figure 6-5, respectively. Specific capital costs for the commercial binary cycle are also plotted on Figure 6-5, and it can be seen that specific capital cost for the 94/6 mixture cycle is about 6% less than the optimum commercial binary cycle.

Table 6-5
Vale Mixed Fluids Case Summary

Case	Wellfield Cost $1000	Condenser Cost $1000	Brine Flow 1000 lb/hr	Specific Output kWh/ 1000 lb	Specific Cost $/kW
300 psia	40,235	21,582	11,056	4.52	2,309
400 psia	36,971	18,440	9,761	5.12	2,245
475 psia	35,206	17,212	9,510	5.26	2,184
525 psia	35,206	16,905	9,454	5.29	2,189

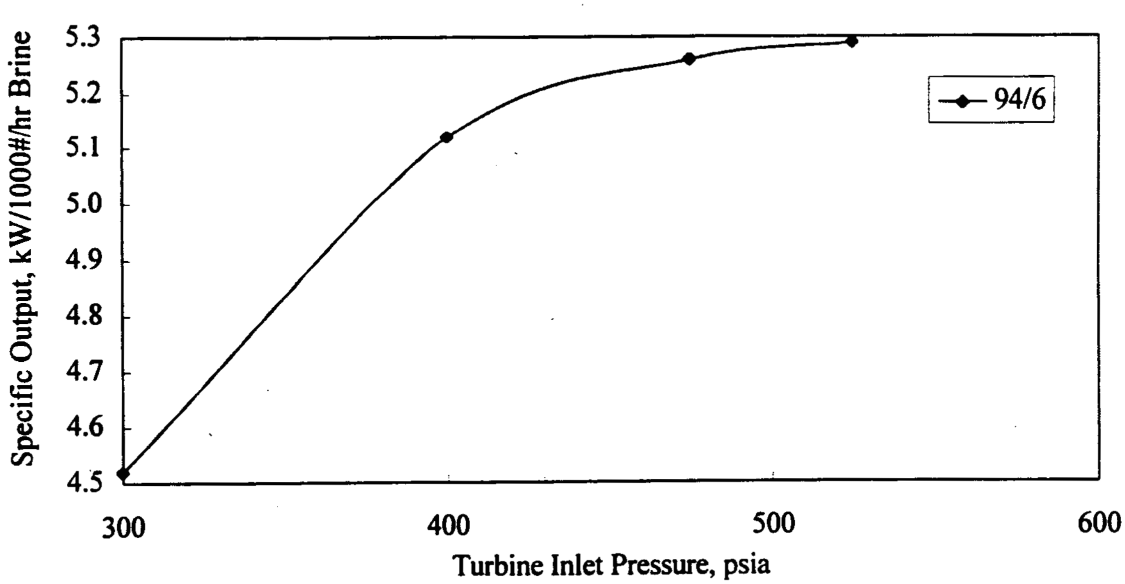

Figure 6-4
Mixed Fluids Specific Output Curve, Vale

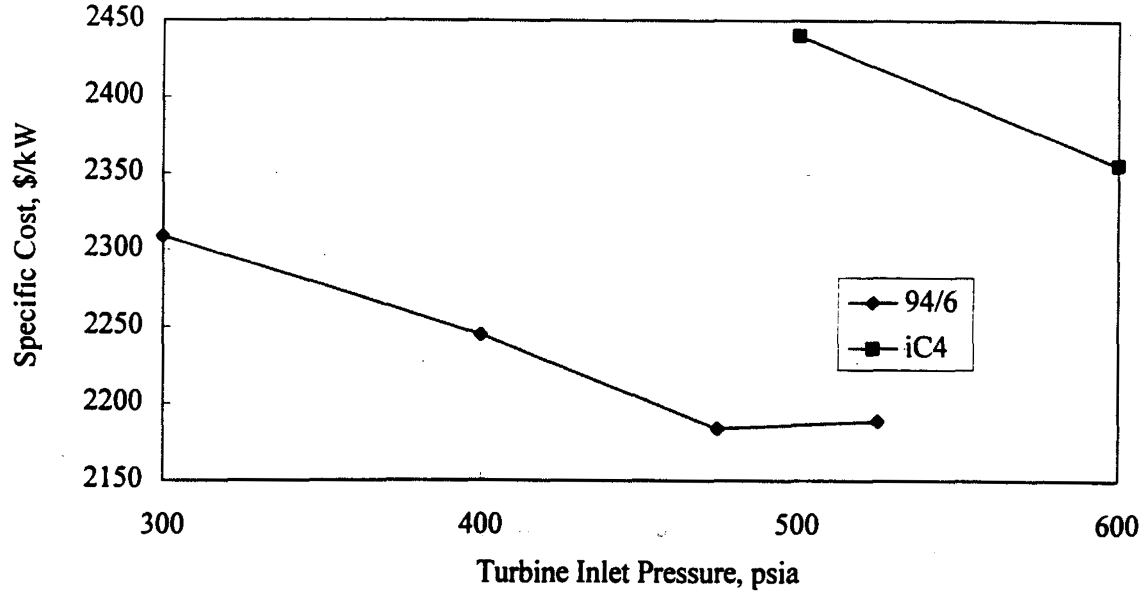

Figure 6-5
Mixed Fluids Specific Capital Cost Curves, Vale

Raft River, Idaho

Three mixtures were used to evaluate the mixed fluids cycles at Raft River (all compositions are on a weight basis):

- 94% isobutane/6% heptane mixture (94/6 mixture)
- 6% propane/88% isobutane/6% heptane (6/88/6 mixture)
- 12% propane/82% isobutane/6% heptane (12/82/6 mixture)

A summary of the results for the three mixtures is presented in Table 6-6 which shows that the 94/6 mixture yields the optimum performance for the mixed fluids binary cycle at Raft River. Figure 6-6 and 6-7 are plots of specific output and specific capital cost, respectively, for the mixed fluids cycles and the commercial binary cycle. It can be observed from these figures that although the specific output of the mixed fluids binary cycle is less than that of the commercial binary cycle, the mixed fluids cycle has a lower capital cost than the commercial binary cycle.

Table 6-6
Raft River Mixed Fluids Case Summary

Case	Wellfield Cost $1000	Condenser Cost $1000	Brine Flow 1000 lb/hr	Specific Output kWh/ 1000 lb	Specific Cost $/kW
12 C3/82 iC4/6 C7:					
250 psia	66,375	31,333	17,512	2.86	3,224
300 psia	57,750	26,732	15,157	3.30	2,961
350 psia	55,875	24,203	14,423	3.47	2,877
400 psia	55,875	22,939	14,929	3.35	2,860
6 C3/88 iC4/6 C7:					
275 psia	55,875	26,443	14,927	3.35	2,891
300 psia	55,875	24,825	14,308	3.49	2,870
325 psia	55,875	23,764	14,410	3.47	2,809
350 psia	55,875	22,865	14,605	3.42	2,797
94 iC4/6 C7:					
200 psia	66,375	30,828	17,145	2.92	3,165
250 psia	55,875	25,514	14,359	3.48	2,837
300 psia	52,500	22,895	14,187	3.52	2,701
350 psia	57,750	21,496	15,115	3.31	2,794

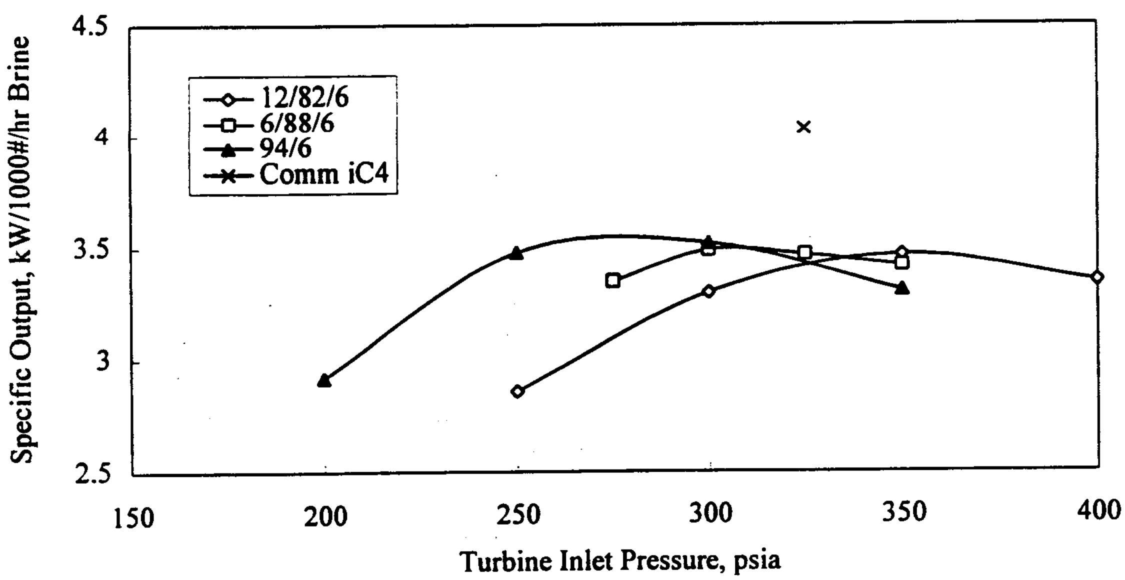

Figure 6-6
Mixed Fluids Specific Output Curves, Raft River

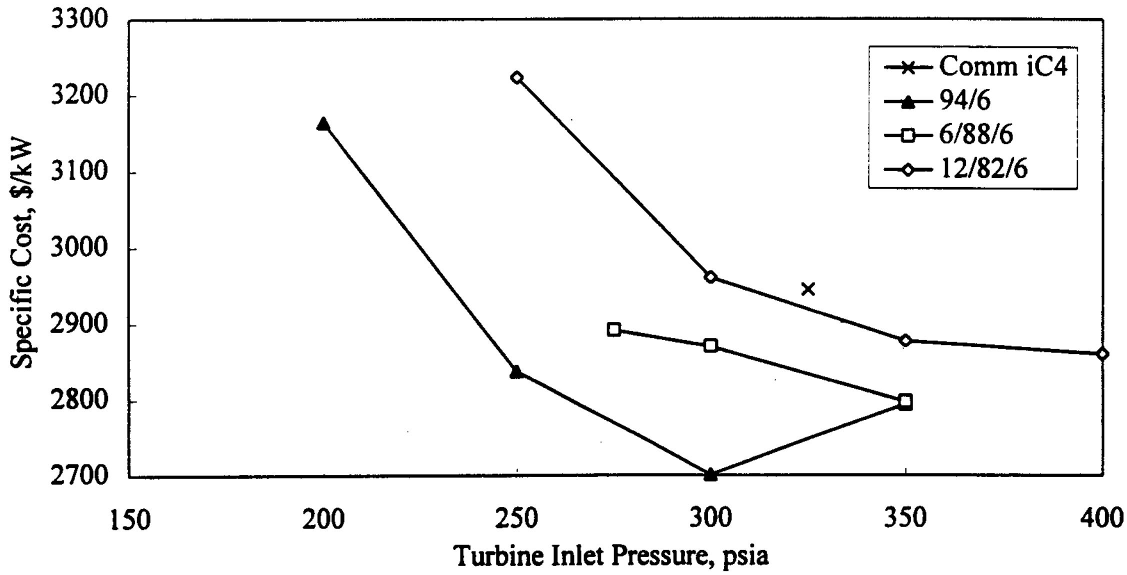

Figure 6-7
Mixed Fluids Specific Capital Cost Curves, Raft River

Thermo Hot Springs, Utah

The effectiveness of the mixed fluids cycle at Thermo Hot Springs was studied using four mixtures as working fluids (all compositions are on a weight basis):

- 6% propane/94% isobutane (6/94 mixture)
- 6% propane/88% isobutane/6% heptane (6/88/6 mixture)
- 12% propane/82% isobutane/6% heptane (12/82/6 mixture)
- 94% isobutane/6% heptane (94/6 mixture)

Based on previous studies (Bliem, 1993) which have indicated that a closer condenser cold end pinch might improve performance for relatively cold resources, mixed fluids cycles with 6/94 mixtures were evaluated for two condensing temperatures: 74°F and 85°F. The results of this evaluation, which are plotted in Figure 6-8, show that cycle performance is better at 74°F condensing temperatures than at 85°F condensing temperatures.

Results of the mixed fluids cycle analysis for Thermo Hot Springs are summarized in Table 6-7. Figures 6-9 and 6-10 plot the specific output and specific capital cost, respectively. Table 6-7 shows that, of the mixtures considered in this study, the 94% isobutane/6% heptane mixture yields the optimum performance for the mixed fluid binary cycle.

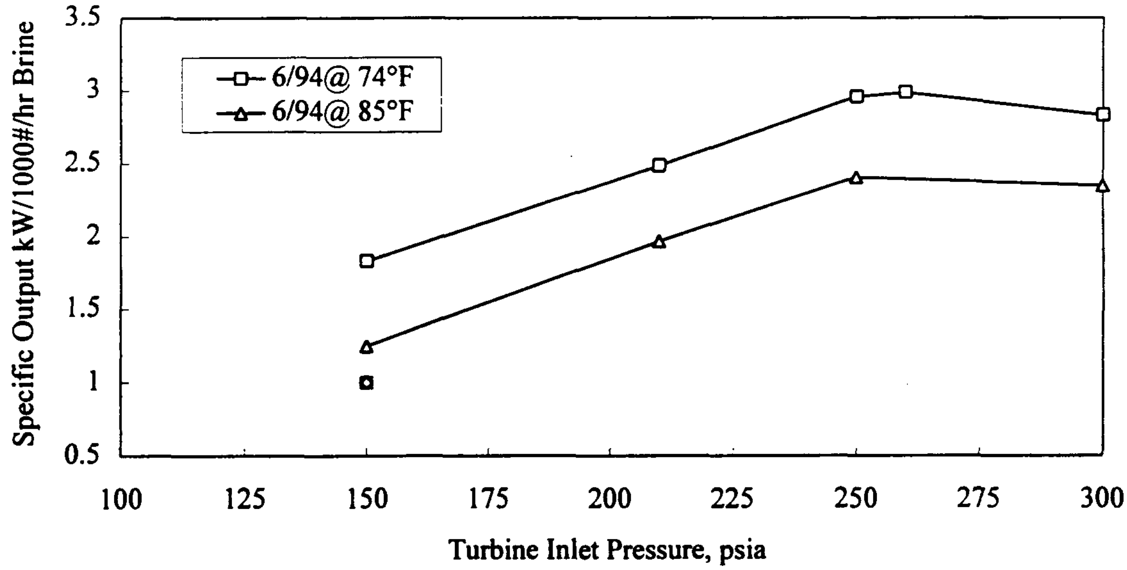

Figure 6-8
Mixed Fluids Optimizing Condensing Temperatures, Thermo Hot Springs

Table 6-7
Thermo Hot Springs Mixed Fluids Case Summary

Case	Wellfield Cost $1000	Condenser Cost $1000	Brine Flow 1000 lb/hr	Specific Output kWh/ 1000 lb	Specific Cost $/kW
6 C3/94 iC4:					
150 psia	145,125	124,716	39,728	1.26	7,403
200 psia	97,875	81,319	26,264	1.90	5,224
250 psia	78,750	64,747	21,171	2.36	4,347
300 psia	82,125	58,314	22,432	2.23	4,340
6 C3/88 iC4/6 C7:					
150 psia	141,750	62,857	39,092	1.28	6,122
200 psia	97,875	41,424	26,213	1.91	4,491
250 psia	87,375	33,998	23,698	2.11	3,978
300 psia	99,750	31,220	27,125	1.84	4,315

Table 6-7 (continued)
Thermo Hot Springs Mixed Fluids Case Summary

Case	Wellfield Cost $1000	Condenser Cost $1000	Brine Flow 1000 lb/hr	Specific Output kWh/ 1000 lb	Specific Cost $/kW
94 iC4/6 C7:					
150 psia	115,500	50,127	31,421	1.59	5,097
200 psia	84,000	35,991	22,909	2.18	3,919
250 psia	89,250	31,204	24,329	2.06	3,966
300 psia	113,625	29,014	31,019	1.61	4,504
12 C3/82 iC4/6 C7:					
150 psia	178,500	78,824	48,832	1.02	7,480
200 psia	110,250	47,489	29,941	1.67	4,965
250 psia	87,375	36,853	23,612	2.12	4,216
300 psia	92,625	32,109	25,179	1.99	4,142

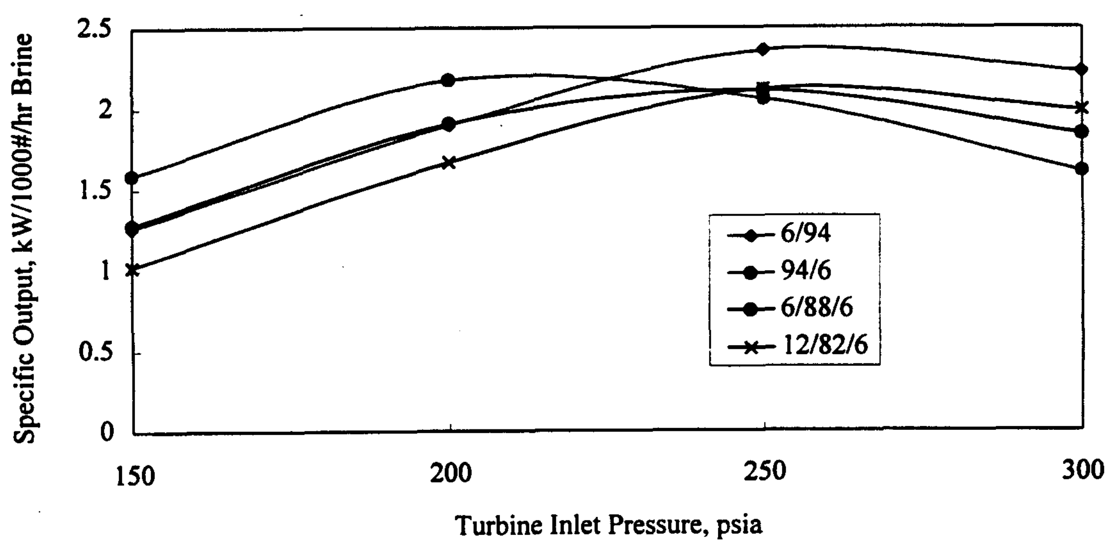

Figure 6-9
Mixed Fluids Specific Output Curves, Thermo Hot Springs

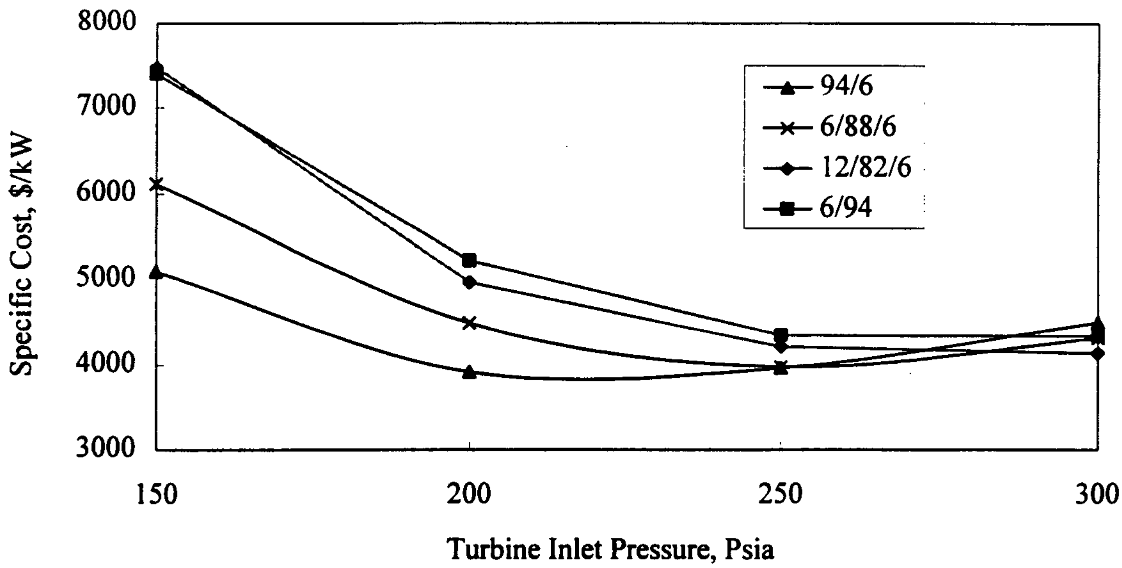

Figure 6-10
Mixed Fluids Specific Capital Cost Curves, Thermo Hot Springs

Synchronous Speed Turbine

Introduction

The synchronous speed turbine is intended to optimize the turbine-generator configuration for the hydrocarbon binary cycles. Although the radial inflow turbine is commonly used in commercial binary power plants, its inherent high speed requires the use of a speed reduction gear between the turbine and the generator. A direct coupled turbine operating at the generator speed (synchronous speed) would avoid the energy losses and cost of the gear box. Small axial flow turbines have been applied to the binary cycle in commercial power plants. This concept, however, anticipates a high efficiency multi-stage turbine with a larger unit size in order to benefit from economies of scale.

Conceptual design of the turbine-generator system was performed by Barber-Nichols, Inc., a co-investigator. From Barber-Nichols' work, we obtained the turbine-generator costs for each resource. These turbine-generator costs were then used to calculate the specific capital cost of power plants for binary cycles utilizing the synchronous speed turbine-generator configurations.

Appendix B contains details of the conceptual design study of the turbine-generator system performed by Barber-Nichols. The turbine-generator configuration proposed by Barber Nichols has the following design features:

- Synchronous turbine-generator system operating at 3600 rpm.
- Axial flow turbine. Barber-Nichols' study found that the performance of multi-stage radial inflow turbines (for 50 MW (net) plant designs) drops off rapidly at relatively low specific speeds. Further, multi-stage radial inflow turbines require large, expensive housings to accommodate the interstage ducting. These two limitations led to the choice of axial flow turbines as the basis for evaluating power plant performance and cost.
- Five stage basic turbine design with differing number of stages to be used for different sites. Rotor configuration and turbine staging details are contained in Appendix B.

Results

Using the turbine-generator cost and performance data from Appendix B, specific capital costs were recalculated for the optimum air-cooled commercial binary cycle power plants and the optimum mixed fluid binary cycle power plants. It may be recalled that the optimum mixed fluid cycle was obtained with 94% isobutane/6% heptane (94/6 mixture) working fluid mixture. These costs are listed in Table 6-8 along with the specific capital costs of the respective optimum power plant designs developed with conventional turbine-generator sets used for commercial binary plants in the other parts of this study. Corresponding specific outputs for the two power plants are also listed in Table 6-8.

Table 6-8
Comparison of Turbines: Specific Cost and Output

Cycle Type	Commercial Binary		Synchronous Speed	
	kWh/1000	$/kW	kWh/1000	$/kW
Commercial Binary				
Glass Mountain	13.05	2,072	13.47	1,726
Surprise Valley	8.49	2,115	8.79	1,817
Vale	5.99	2,356	5.97	2,346
Raft River	4.04	2,945	4.17	2,841
Thermo Hot Springs	2.44	4,188	2.58	3,924
Mixed Fluids (94 iC4/6 C7)				
Surprise Valley	7.55	1,839	8.10	1,770
Vale	5.26	2,184	5.57	2,072
Raft River	3.52	2,701	3.78	2,541
Thermo Hot Springs	2.18	3,919	2.25	3,633

Table 6-8 shows that the use of the lower cost Barber-Nichols turbine-generator sets results in lower specific capital costs compared to power plants that use conventional turbines. Synchronous speed turbine-generator sets would not only be less expensive than conventional turbine-generator sets but would also be more efficient. Hence, the use of such a generator would also improve specific output as can be seen from Table 6-8.

Metastable Expansion

Introduction

Commercial binary plants are usually designed with enough superheat in the turbine inlet vapor so that the subsequent expansion remains entirely in the vapor region. Metastable expansion technology aims to improve specific output of super-critical cycles by minimizing superheat in the turbine inlet vapor, and expanding the working fluid in the turbine through the two phase region. Expansion in the turbine through the two phase region is intended to take place at a rapid rate so that liquid droplets are not formed and the expansion is metastable. Working fluid enters the turbine as vapor, passes into, through and exits the two phase region while still in the turbine nozzles, and leaves the turbine as superheated vapor. If turbine efficiency does not suffer during metastable expansion, other work has shown that specific output could be increased by the use of metastable expansion. (Mines, 1994a)

Experimental studies have demonstrated that metastable expansion can be obtained during isentropic expansion in the two phase region. (Mines, 1994b)

Metastable expansion technology is applied near the critical point of the working fluid, a region of phase equilibria characterized by uncertainty in data. Consequently, different thermodynamic models in the critical point region can give substantially different results. Therefore, for a meaningful assessment of metastable expansion technology to emerge, the relative effect of model differences must be understood. Towards that end, calculations were made to compare with values reported in the literature. These calculations were made using the assumptions in the literature rather than the assumptions used in this study.

Cycle Process Flow

Metastable expansion technology is applicable to standard binary cycles and the reader is referred to Section 5 for a typical process flow diagram and description.

Literature Comparison

Performance Analysis. Figure 6-11 replots the literature data for a hypothetical isobutane binary cycle at 550 psia and a 10°F heat exchanger approach, for a 335°F resource (Mines, 1994c). Using a turbine inlet temperature of 290°F as the reference, the figure shows the incremental effect on specific output, hydrocarbon/brine ratio, and turbine enthalpy drop (ΔH) as the turbine inlet temperature is reduced from 290°F to 280°F for a constant plant output. The figure also shows results for the same calculations obtained with the thermodynamic model used in this study. A comparison of the two sets of data reveals that although the data are directionally similar, differences in magnitude exist. For example, the incremental improvement in specific output calculated using this study's model is only about half of that reported by Mines.

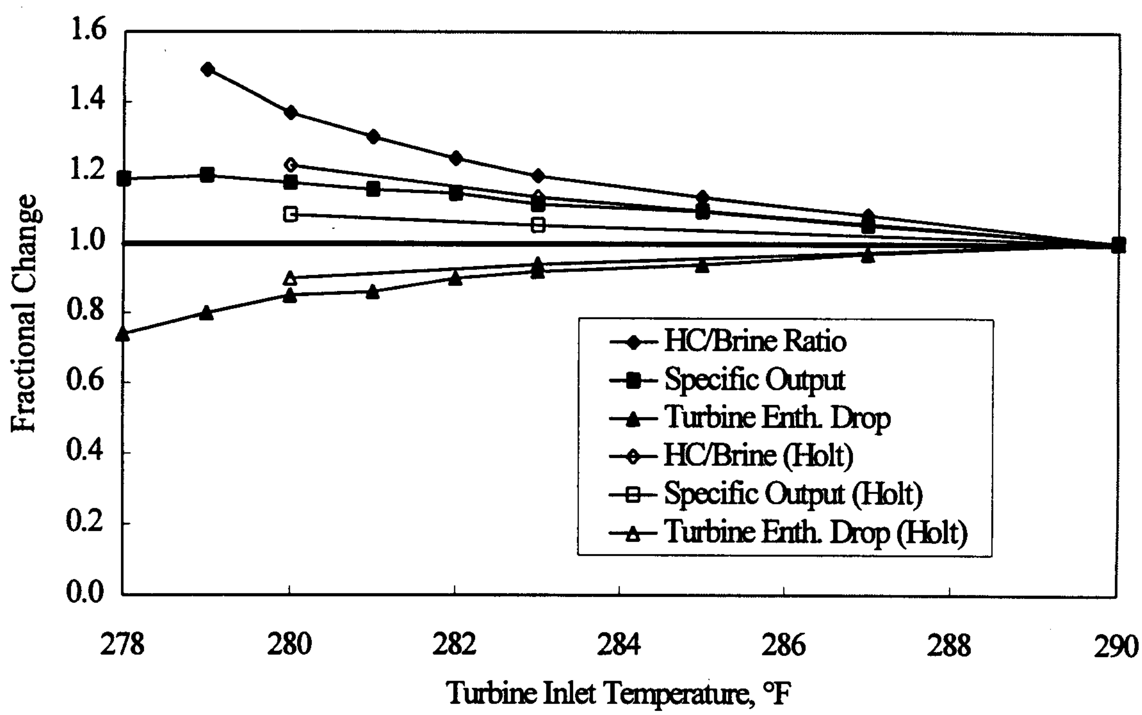

Figure 6-11
Metastable Expansion for Hypothetical Cycle

The differences between the two sets of data can largely be ascribed to differences in the respective equations of state and composition of the working fluids. The Mines values were calculated using the NIST-12 model for pure isobutane; this study used the Starling equation of state for commercial isobutane. In order to identify the differences in these models, the two equations of state were used to prepare a plot of temperature versus enthalpy for pure isobutane, Figure 6-12. Figure 6-12 also shows the temperature-enthalpy plot for commercial isobutane calculated using the Starling equation of state. It is evident from the figure that the predictions of the two equations of state diverge near the critical point (273°F) (the region used for metastable expansion). In the region of the critical point, enthalpy differences as large as 12 Btu/lb hydrocarbon occur between NIST with pure isobutane and Starling with commercial isobutane.

The question as to which equation of state is correct is an important question but one that is beyond the scope of this study. There are few reliable data available to validate the models, and the uncertainty surrounding calculations in the critical point region will not be eliminated without further research.

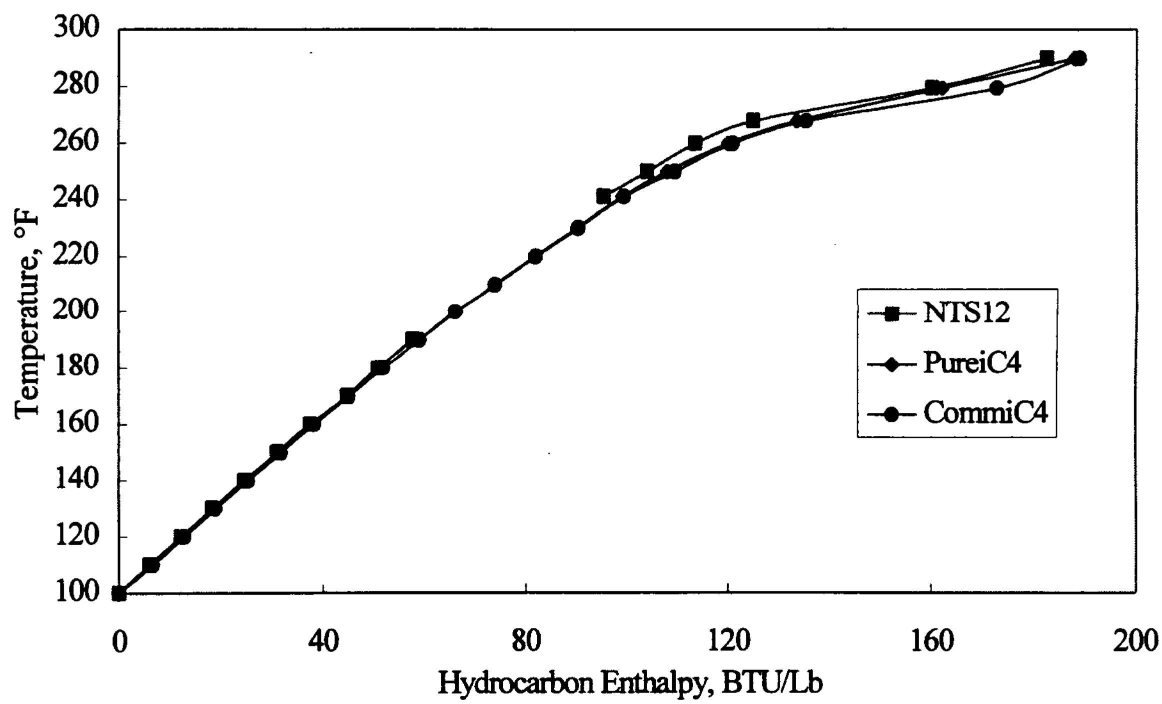

Figure 6-12
Comparison of NIST-12 and Starling Equation of State

Cost Analysis. Using a hypothetical, isobutane binary cycle it has been shown that binary cycles with metastable expansion can yield higher specific output than commercial (non-metastable exapansion) binary cycles (Mines, 1993) if the brine injection temperature is not constrained. This improvement in specific output was confirmed in this study. However, for that hypothetical cycle (550 psia, 10°F pinch, 335°F resource), the higher specific output with metastable expansion did not result in a power plant with a lower specific capital cost compared to a power plant based on the commercial air-cooled binary cycle.

Figure 6-13 plots the ratio of specific capital costs for the metastable binary cycle to the commercial binary cycle. As can be seen from the figure, the metastable power plant cost is up to 2% more than the commercial binary power plant cost. The metastable expansion binary power plant has a higher capital cost due to increased hydrocarbon circulation, higher heat exchanger costs due to lower LMTDs, and larger size and cost of turbine-generator sets due to lower turbine enthalpy drops. Table 6-9 provides a comparative equipment cost summary for the metastable and commercial binary cycles for the above mentioned hypothetical case.

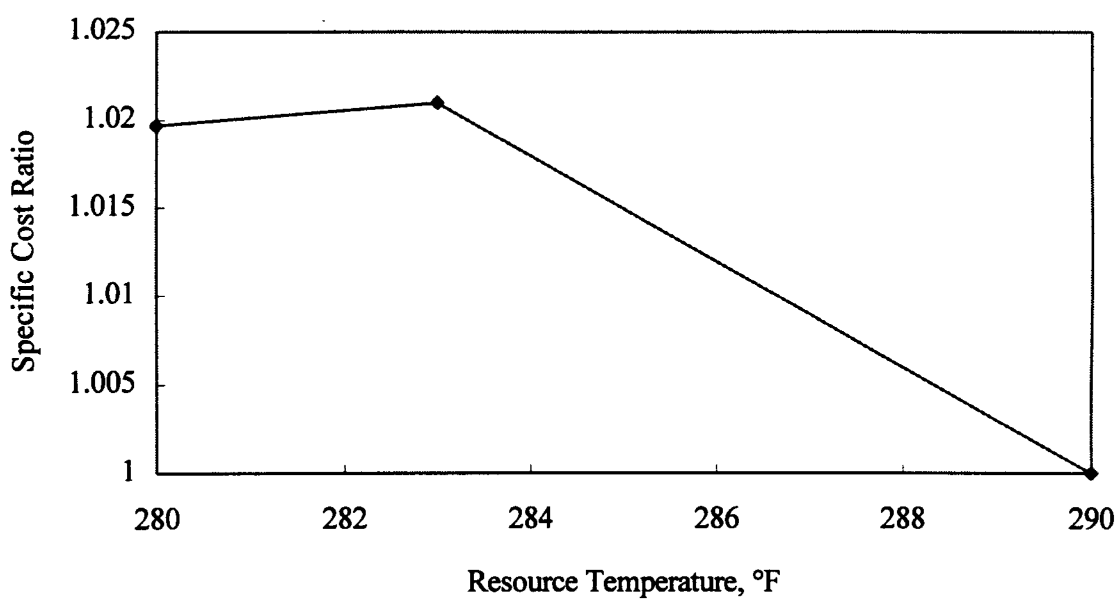

Figure 6-13
Ratio of Metastable Expansion and Commercial Binary Costs

Table 6-9
Metastable Expansion: Comparison of Equipment Costs

		Base Case @290°F	Metastable @ 283°F
Brine Utilization, kWh/1000 #		5.99	6.35
Brine/Isobutane Heat Exchanger	Duty	1,528	1,546
	LMTD	20.3	18.8
	Cost, $1000	4,901	5,634
Air Condenser	Duty	1,311	1,331
	LMTD	24.1	23.5
	Cost, $1000	10,879	11,035
Turbine Generator Set	Cost, $1000	11,542	11,642
Well Pumps	Cost, $1000	1,791	1,639

Performance Analysis. Binary cycles with metastable expansion were analyzed using the same methodology that was used to model the commercial binary cycles in Section 5. Cycle thermodynamics were modeled using the Starling equation of state. Although this study's evaluation of metastable expansion technology might differ from an evaluation obtained using a different equation of state, for example NIST-12, our approach allows a meaningful comparison of metastable expansion technology with other binary technologies.

In order to determine the specific output and cost per kilowatt of electricity, binary cycles using metastable expansion were applied to three sites: Surprise Valley, Vale, and Raft River. The resource at Thermo Hot Springs is too cold for a supercritical cycle based on isobutane and was therefore not considered.

In general, the assumptions used for simulating commercial binary cycles were also used to simulate binary cycles with metastable expansion. Two key assumptions used in the development of binary cycles with metastable expansion need to be highlighted.

- First, a turbine efficiency of 85% with no loss of turbine efficiency during the expansion through the two phase region was assumed. Although the vapor is thought to remain in a supercritical condition as it passes through the turbine, recent test have shown signs of erosion within the inlet nozzles (Mines, 1994). However, at the time of this writing the effect of nozzle erosion on turbine effieciency is not clear.

- Second, the brine injection temperature was limited to a minimum of 150°F, the minimum injection temperature specified by EPRI for the three sites considered for metastable expansion. In our literature comparison study, brine outlet temperature was allowed to decrease at lower turbine inlet temperatures.

Cost Analysis. The cost analysis methodology for binary cycles with metastable expansion was identical to that used for cost analysis of commercial air-cooled binary cycles as described in Section 5.

Results

Specific output curves for the metastable binary cycles evaluated in this study are plotted in Figures 6-14, 6-15 and 6-16 for Vale, Surprise Valley, and Raft River, respectively. Specific output for the respective commercial binary cycles are also shown on the figures. It is evident that the optimum commercial binary cycles yield a higher specific output compared to the corresponding metastable expansion cycles. The results plotted in Figures 6-14, 6-15 and 6-16 were obtained by constraining the brine injection temperature to be at least 150°F. This constraint imposed on all technologies evaluated in this study may explain why the specific output of binary cycles with metastable expansion was lower than the specific output of the optimum commercial binary cycles.

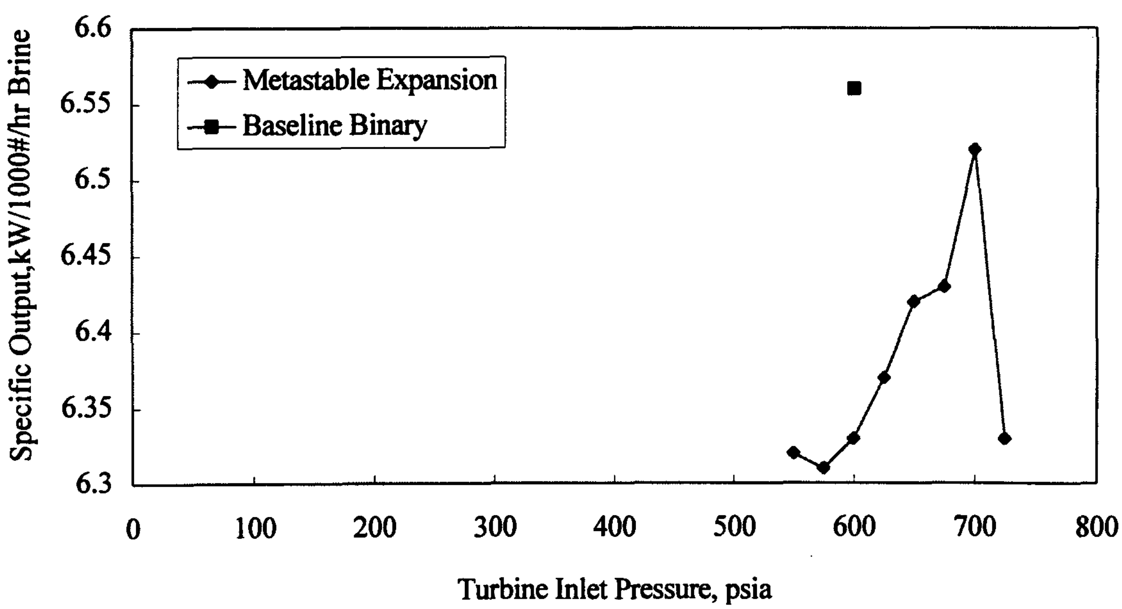

Figure 6-14
Metastable Expansion Binary Specific Output Curves, Vale

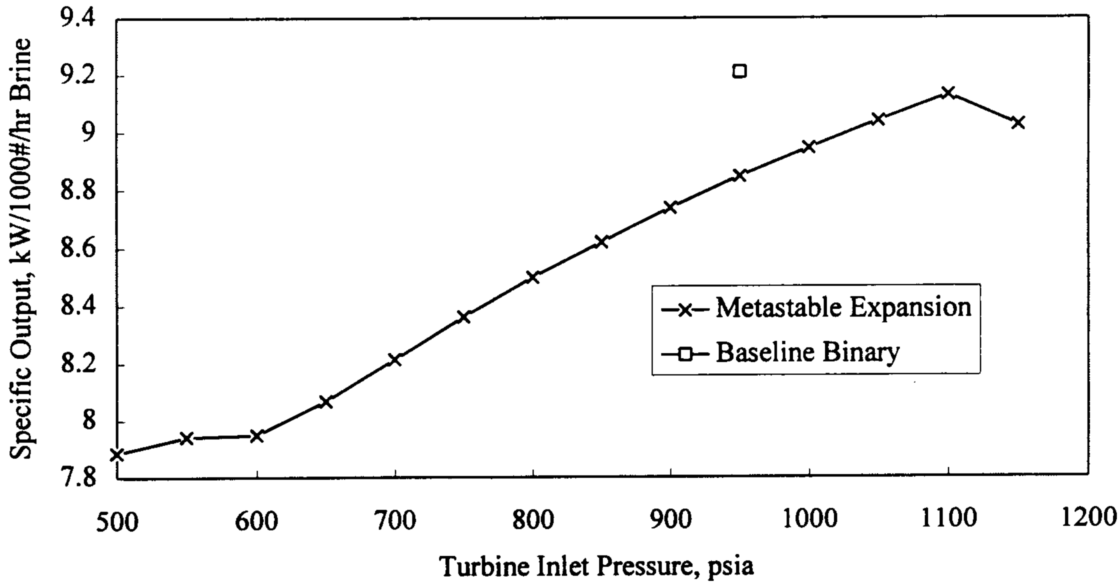

Figure 6-15
Metastable Expansion Binary Specific Output Curves, Surprise Valley

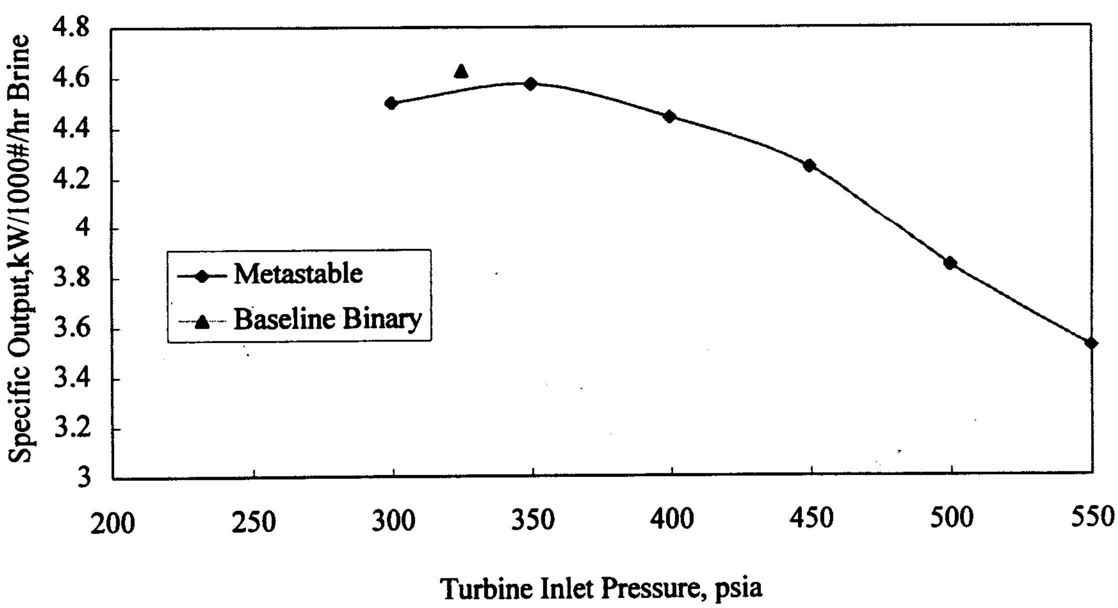

Figure 6-16
Metastable Expansion Binary Specific Output Curves, Raft River

Table 6-10 summarizes the performance of metastable expansion binary cycles; the table also lists the specific capital cost for the optimum commercial binary cycles. For two of the sites, the baseline binary cycle has a slightly lower specific capital cost and for the third (Raft River), metastable expansion is slightly lower. It is clear that metastable expansion does not offer a dramatic improvement over current technology. The table also shows that the metastable expansion binary cycle that yields the maximum specific output does not correspond to the lowest specific capital cost for the three resources considered in this study.

Table 6-10
Metastable Expansion: Summary of Cases

	Metastable Expansion				Commercial Binary	
	Maximum Specific Output Case		Economic Optimum Case		Economic Optimum Case	
Resource	Turbine Inlet	Cost ($/kW)	Turbine Inlet	Cost ($/kW)	Turbine Inlet	Cost ($/kW)
Vale	700 psia, 306.9°F	2,486	600 psia, 289.6°F	2,464	610 psia, 293.5°F	2,356
Surprise Val.	1050 psia, 353°F	2,217	500 psia, 269.6°F	2,157	850 psia, 353°F	2,115
Raft River	450 psia, 255°F	3,024	350 psia, 228.4°F	2,925	325 psia, 221°F	2,945

Binary Cycle: Hot Dry Rocks

Introduction

Clear Lake, Geysers is the hot dry rock resource analyzed in this study. The temperature of this resource per Table 4-1 is 380°F which is almost equal to the Surprise Valley resource temperature. Therefore, for the purposes of this study hot dry rock technology was evaluated using binary cycles optimized for Surprise Valley. Thus, the hot brine temperature and injection temperature were assumed to be 375°F and 150°F, respectively. Further, only binary cycles were used to design the hot dry rock power plants because at Surprise Valley in both specific output and specific capital cost binary cycles were superior to flash cycles (Figures 5-15 and 5-16).

Cycle Process Flow

A schematic of one possible well injection and production concept is shown in Figure 6-17 (Duchane, 1993).

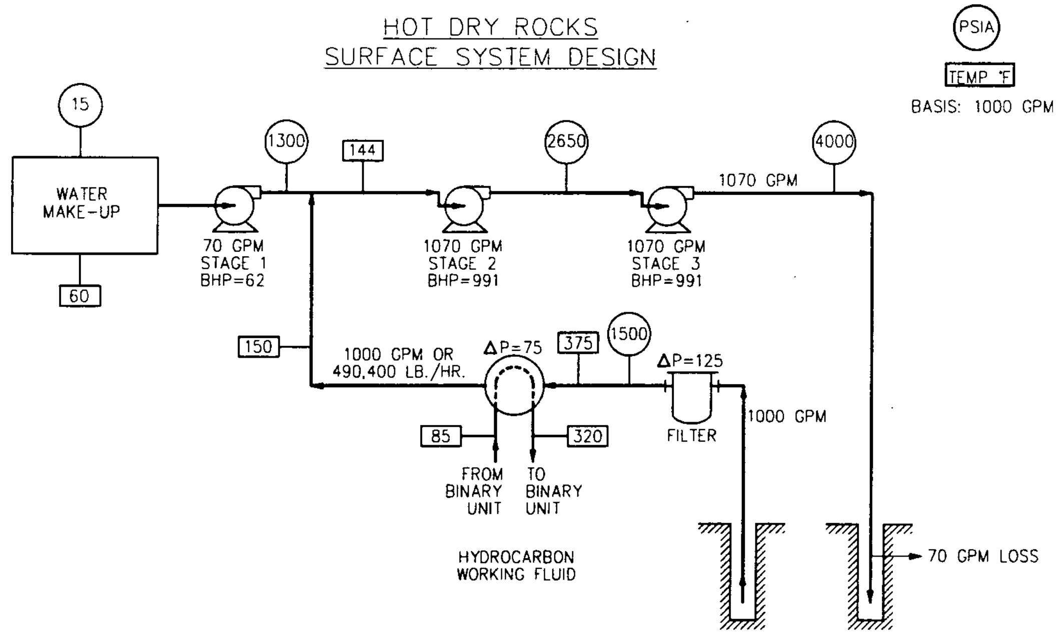

Figure 6-17
Hot Dry Rocks Surface System Design

Performance Analysis. Basic parameters including the 4,000 psia injection pressure, well flow of 200,000 lb/h and 1500 psia production well pressure were obtained from Table 4-1. Heated brine from the hot dry rock system was fed to a binary cycle. Two types of binary cycles were used: a standard commercial binary cycle unit at 500 PSIA and 277°F and a mixed fluids (94% IC4/6% C7) cycle.

To estimate water losses that will occur upon injection and production of the brine, Mr. Dave Duchane of the Los Alamos Laboratory was consulted. It appears that water losses are currently being estimated in the range of 7% of total water flow rate (Duchane, 1994).

Economic Analysis. The binary cycle cost model was modified to include the features of the hot dry rock cycle developed at Los Alamos. For example, the cost model was modified to include three injection pumps in series to achieve the required 4000 psia injection pressure. It was also modified to adjust the brine/hydrocarbon heat exchangers for the high tubeside pressures that would be encountered with the high pressure brine. Well drilling costs of $6,000,000 (including fracturing) for 200,000 lb/hr wells were used to develop the well field costs.

Results

A 50 MW plant using a standard 500 psia binary cycle would cost $9,500/kW, and a 50 MW plant using 94/6 mixed fluids would cost $8,900/kW. These specific capital costs are much higher than those for the other binary units. Two significant factors are largely responsible for the high costs of power plants that would use hot dry rocks as a source of energy: (1) the high well field parasitic load which consumes 30% of the gross power production, and (2) the $6,000,000 to drill and fracture each injection/production well pair.

Moreover, the flow per well is only 200,000 lbs/hr per well which leads to a large number of wells being required. The wellfield cost, $300 million for the cases considered here, thus dominates the total capital cost.

7

ADVANCED FLASH CYCLES

Dual Flash/Rotary Separator Turbine

Introduction

Dual flash/rotary separator turbine (Dual Flash/RST) technology is an advanced flash cycle that has been under development for several years. In contrast to a conventional dual flash plant in which the geothermal brine is flashed isenthalpically across a throttle valve to produce high pressure steam, in a plant that uses rotary separator turbine (RST) technology, the brine is flashed across a two-phase nozzle. The entropy change across a two-phase nozzle is less than that across a throttle valve. As a result, less available energy is lost in the rotary separator turbine compared to the available energy lost across the throttle valve in a dual flash plant (Cerini and Hays, 1980).

Cycle Process Flow

Figure 7-1 is the process flow diagram for a dual flash plant with a rotary separator turbine downstream of the production well. The rotary separator turbine replaces the high pressure separator in a conventional dual flash plant, and it performs the functions of flashing the geothermal brine, and separating liquid brine from high pressure steam. It also generates power from the two phase stream in the process of separating it.

In essence, the rotary separator turbine combines a liquid impulse turbine with an axial steam turbine on the same shaft. The impulse turbine extracts work from liquid brine and the axial turbine extracts work from steam. Steam from the RST then flows to a standard geothermal steam turbine and the brine is sent to the low pressure separator. The remaining process steps are the same as those in a conventional dual flash plant.

Performance Analysis

Dual flash/RST technology was evaluated for Desert Peak, Dixie Valley, Glass Mountain, Coso, and Salton Sea. At the other sites dual flash cycles (the predominant component of dual flash/RST technology) are inferior to binary cycles. Also, RST efficiencies are low at low resource enthalpies.

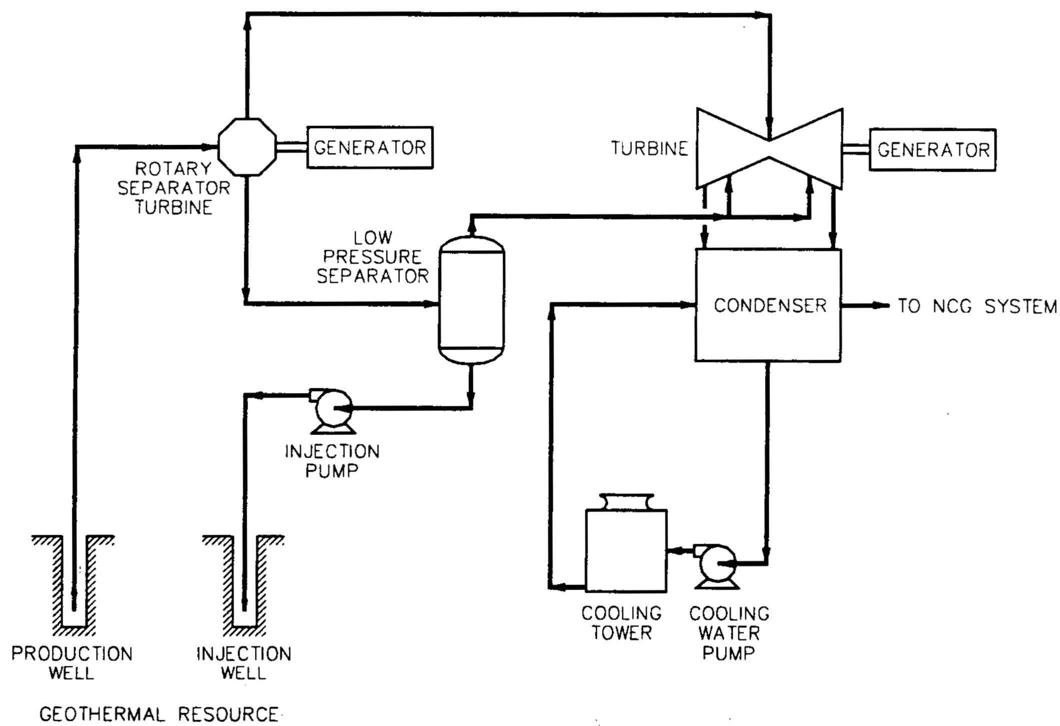

Figure 7-1
Dual Flash/Rotary Separator Turbine: Process Flow Diagram

Description. In a geothermal power plant utilizing dual flash/rotary separator turbine technology, the plant downstream of the rotary separator turbine (RST) is a conventional dual flash plant, as noted above. Thus, the modeling assumptions and methodology that apply to a dual flash plant also apply to the dual flash section of plant based on dual flash/RST technology.

The power output from the RST was calculated by using efficiency data furnished by Douglas Energy, the RST developer. Consistency was confirmed by comparing model predictions with published experimental RST data (Cerini, 1978). The overall efficiency of the RST is a function of inlet pressure, outlet pressure, and inlet vapor fraction. Turbine efficiency increases with an increase in each of these parameters, and inlet vapor fraction is the most significant of these parameters. An interpolation routine to calculate the overall efficiency of the RST using these parameters was added to the Holt dual flash model.

Assumptions. In addition to the general assumptions that were used to analyze dual flash power plants, the following specific assumptions were used to evaluate RST technology:

- Generator efficiency of 95%.
- Wellhead pressures listed in Table 7-1. All wells at a given site have the same wellhead pressures and flow rates.

Table 7-1
Dual Flash/RST Cycle Process Summary

Site	Well P (RST Inlet P) (psia)	Flash P (RST Outlet P) (psia)	Dual Fl High Flash (psia)	Brine Flow Rate (lb / hr)	Well Steam Quality (%)	RST Overall Eff (%)	Gross RST Power (kW)	Net Power v. Dual Fl (kW)	Spec Capital Cost ($/kW)	DF Spec Capital Cost ($/kW)
Desert Peak	90.0	71.0	90.0	4,500,000	12.30	26.50	883	242	1,513	1,491
Dixie Valley	100.0	80.0	100.0	4,000,000	14.65	30.59	1,032	331	1,235	1,225
Glass Mtn	151.0	100.0	151.0	3,000,000	19.43	39.68	2,456	572	1,523	1,504
Coso Hot Sp	130.0	100.0	130.0	3,200,000	22.69	40.46	2,000	418	1,608	1,588
Salton Sea	325.0	118.0	151.0	2,600,000	21.65	46.57	6,058	3,045	1,083	1,057

Cost Analysis

Description. Overall cost methodology used to analyze dual flash/RST technology was the same as that used for dual flash power plants. A Douglas Energy estimate of $472/kW for a 3 MW unit was used as the basis for RST costs. This estimate was multiplied by 1.8 to obtain the installed cost of the 3 MW unit. The installation factor of 1.8 (compared to 2.53 for a steam turbine) was obtained by summing up direct and indirect costs. Indirect costs were calculated as a percentage of direct costs.

Assumptions. Specific assumptions used in the cost evaluation of dual flash/RST power plants were:

- A minimum price of $250,000 for 0.53 MW units or smaller.
- At least one RST per well pad.

Results

Performance. This study confirms that a dual flash/RST plant will yield a higher specific output than a conventional dual flash plant. The percent increase in specific output of a dual flash/RST plant over a conventional dual flash plant is plotted in Figure 7-2. For all the sites other than Salton Sea, specific output of dual flash/RST plants is only about 1 percent better than conventional dual flash plants. For these sites the wellhead pressures are already at or below the optimum high pressure flash pressure, and therefore the resources at these sites do not have to be throttled. Therefore the advantage that accrues from the more efficient expansion process in an RST is negated. Somewhat higher power is generated using the RST because the RST liquid turbine generates power from the liquid brine stream.

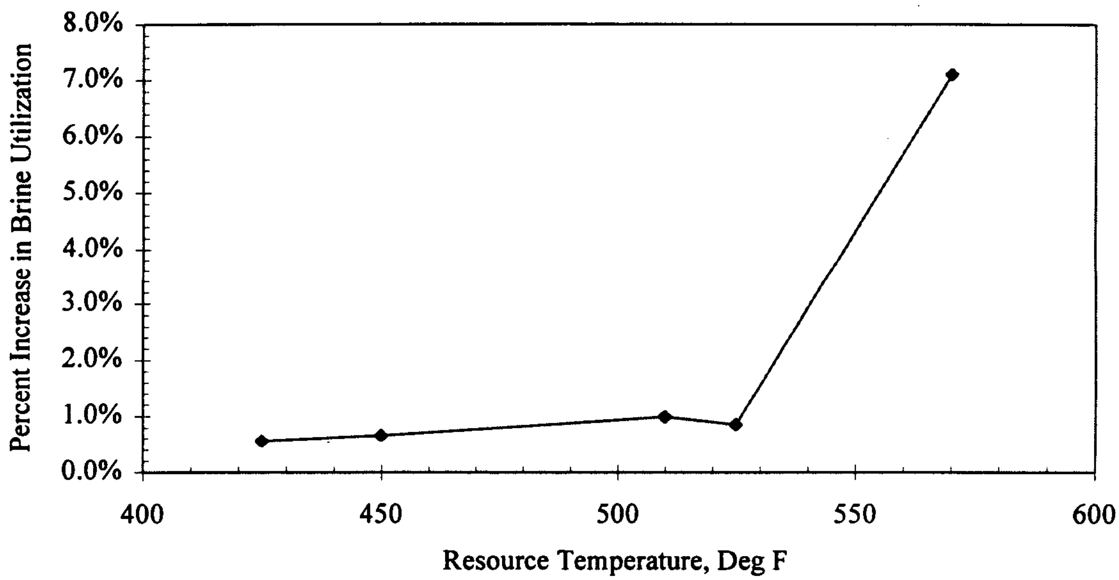

Figure 7-2
RST: Incremental Change in Specific Output v. Commercial Dual Flash

A process summary for the maximum specific output dual flash/RST plants is presented in Table 7-1 which confirms that only for Salton Sea is the wellhead pressure significantly higher than the optimum high pressure flash pressure for a dual flash plant. Wellhead pressure at Salton Sea is 325 psi which is almost 175 psi higher than the inlet pressure to the geothermal steam turbine since turbine inlet pressures are limited to 153 psia for steam turbines used in geothermal power plants for metallurgical reasons. Thus, the RST uses the additional 175 psi drop to produce power while the dual flash plant merely throttles the stream. As a result, specific output a of dual flash/RST plant at Salton Sea would be 7% higher than that of a conventional dual flash power plant.

An important assumption of this study is that wellhead pressures and flows are the same for all wells at a given site. For resources which have a significant variation in wellhead pressures, the RST technology may be an option to improve the overall performance of a steam flash plant by recovering energy that would otherwise be lost in throttling the steam to the lowest common pressure. Currently, the Department of Energy is sponsoring a test of RST at Coso on a well that has a significantly higher pressure than other wells feeding the high pressure steam manifold (Hayes, 1994).

It appears that RST technology would be beneficial in improving specific output for resources in which the wellhead pressure exceeds the allowable turbine inlet pressure because of technological or other constraints.

Economics. The specific capital cost for dual flash/RST plants is higher than the specific capital cost for standard dual flash plants as can be seen from Table 7-1. This finding can be explained by using Salton Sea as an example.

Figure 7-3 is a plot of the specific capital cost versus flash pressure for a dual flash/RST power plant at Salton Sea. The specific capital cost of the optimum dual flash plant is also shown as a point on the figure. It is evident that specific

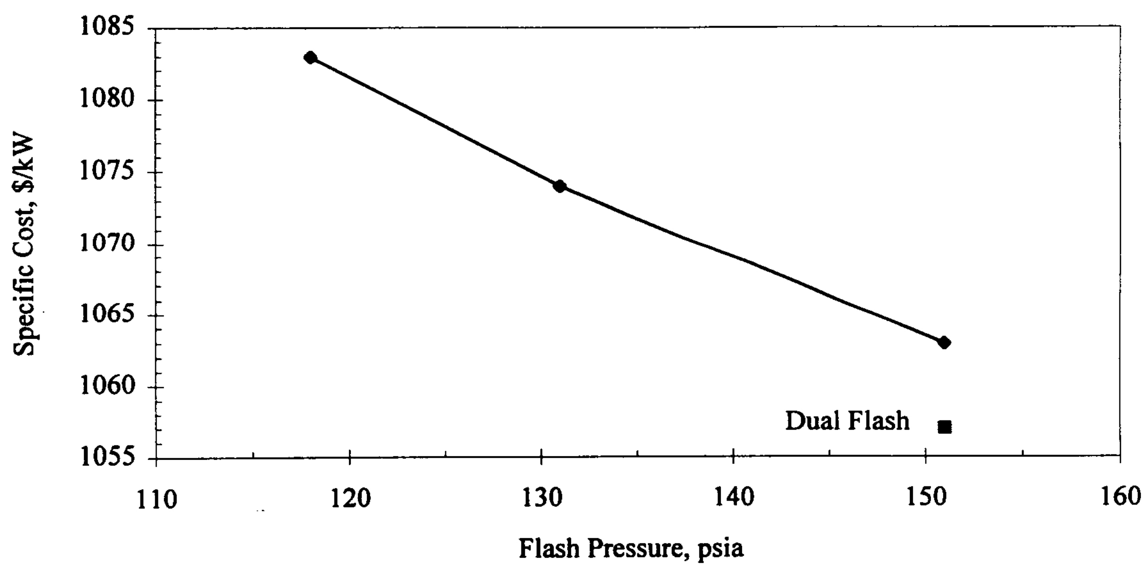

Figure 7-3
RST: Specific Capital Cost v. Flash Pressure, Salton Sea

capital cost of a conventional dual flash plant is lower than a dual flash/RST plant. Dual flash/RST plants are likely to have a higher specific capital cost than conventional dual flash power plants because of the cost differential between an RST and a steam turbine.

The installed cost of an RST is $876/kW whereas the installed cost for a steam turbine is $585/kW although the installation factor for the RST was 1.8 compared to 2.53 for the steam turbine. The increased specific output that the RST yields is generally not sufficient to offset this cost differential. In fact, the highest specific capital cost point on Figure 7-3 corresponds to the highest specific output.

It would appear from Figure 7-3 that raising the flash pressure would lower the specific capital cost. However, it should be remembered that the flash (turbine inlet) pressure for both the RST and conventional dual flash plants is limited to 153 psia due to metallurgical constraints on the geothermal steam turbine. Removing this constraint would result in lower specific capital costs for the dual flash/RST plant but specific capital costs of conventional dual flash plants would also be lowered.

Dual Flash/Steam Reheater

Introduction

The addition of a steam reheater to the dual flash process has been proposed to improve cycle performance (Li and Priddy, 1985). This concept involves using a steam reheater to superheat the exhaust from the high pressure turbine stages with heat from the saturated liquid from the high pressure separator. This technology has found some application in advanced nuclear power plants (Li and Priddy, 1985). A theoretical investigation of the use of a steam reheater for geothermal applications has been reported (DiPippo and Vrane, 1991). The authors concluded that addition of a steam reheater can yield from one to six percent more work for the same brine flow rate compared to a conventional dual flash plant.

Cycle Process Flow

Figure 7-4 shows a process flow diagram for a dual flash/steam reheater geothermal power plant. The plant is a modification of a commercial dual flash plant. In a commercial dual flash plant, the brine from the high pressure separator is flashed in a low pressure separator to generate low pressure steam. In a dual flash/steam reheater plant, the brine from the high pressure separator is used to superheat the outlet from the high pressure turbine blades. This superheated steam is then mixed isobarically with the saturated steam from the low pressure separator. The combined stream is then sent to the low pressure turbine blades to produce the remainder of the power. The increase in power output is due to increased efficiency in the low pressure section because the

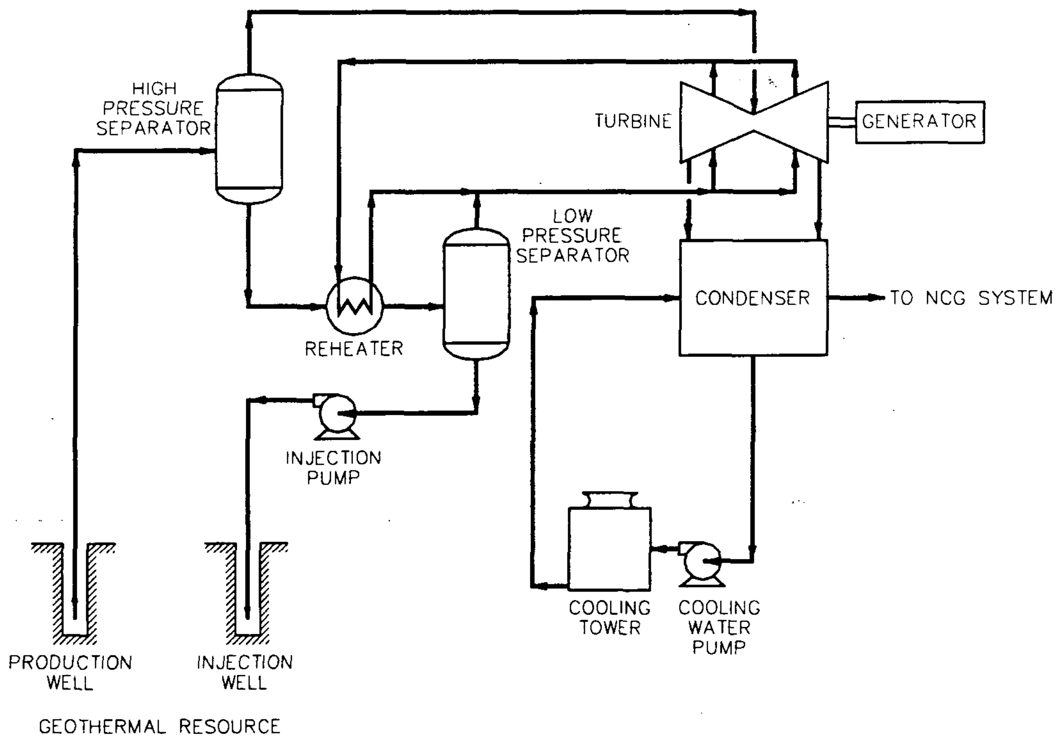

Figure 7-4
Dual Flash/Steam Reheater: Process Flow Diagram

superheated low pressure steam leaves the exhaust with less moisture content that if it were initially saturated as in a standard dual flash cycle.

Performance Analysis

Description. The optimization approach used for the analysis of dual flash plants was also used for analyzing the performance of power plants based on dual flash/steam reheater technology. Thus, the optimization process involves calculating optimal values of condenser approach temperature, cooling water flow rate, etc.

Assumptions. The following assumptions were used to analyze the performance of dual flash/steam reheater plants:

- The increased efficiency of the low pressure turbine section is accounted for using the Bauman rule (Bauman, 1921). The efficiency of the low pressure turbine section in a commercial dual flash plant is 85%. For dual flash/steam reheater plants the efficiency is increased by 1% over the baseline value of 85% for every 2% reduction in the moisture content of the exhaust steam.

- A 2 psi pressure drop is assumed for the steam side of the steam reheater.
- Reheater approach temperature is 5°F, overall heat transfer coefficient is 40 Btu/h-ft^2-°F and cost $8.50 per square foot (bare surface). These valves were determined using HTC-STX, a commercially available computer program for designing and rating heat exchangers.

Cost Analysis

Description. The basic cost model used to analyze commercial dual flash plants was modified to analyze dual flash/steam reheater plants to account for the cost of the steam reheater. The steam reheater is a shell and tube heat exchanger. Its duty and log mean temperature difference (LMTD) were calculated using the commercial process simulator HYSIM for given stream inlet conditions.

Results

Figure 7-5 which plots the percent increase in specific output of plants with steam reheaters over conventional dual flash plants shows that for the colder resources, employing a steam reheater actually reduces brine utilization. This is due to the pressure drop in the reheater and associated piping which reduces the second law efficiency of the cycle. For the colder resources, this drop in second law efficiency offsets any gains in turbine efficiency that the steam reheater

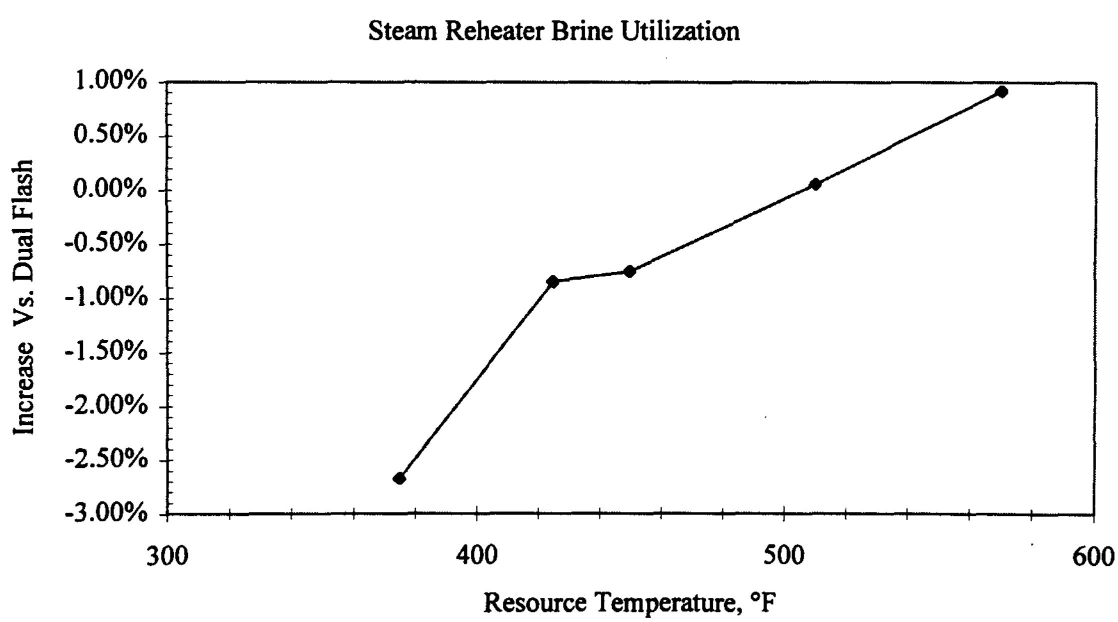

Figure 7-5
Dual Flash/Steam Reheater: Incremental Change in Specific Power Output

provides compared to the dual flash cycle. For Salton Sea and Glass Mountain, the steam reheater pressure drop is a smaller fraction of the total cycle pressure drop and the increase in blade efficiency is enough to overcome the decrease due to the pressure drop. Therefore there is a net increase in specific output.

Tables 7-2 and 7-3 summarize the performance of the dual flash/steam reheater power plants for Surprise Valley, Desert Peak, Dixie Valley, Glass Mountain, and Salton Sea. Specific capital cost for the various sites are also presented in Table 7-3 along with the specific capital cost for dual flash power plants at the respective sites. It can be observed from Table 7-3 that the flash plants with steam reheater have higher specific capital costs than conventional dual flash plants for all the sites listed in the table. Clearly, on an economic basis increased capital costs associated with the steam reheater do not justify the increased specific output that a steam reheater can provide.

Table 7-2
Dual Flash/Steam Reheater Process Parameters

Site	Res. Temp (F)	High Press Steam (M lb/hr)	Low Press Steam (M lb/hr)	Turb Press High (psia)	Turb Press Low (psia)	Cond Press (in Hg)	Turbine Quality w/o Reheater	Turbine Exhaust Quality w/ Reheater	Low Press Blade Eff.	High P Gross Power (kW)	Low P Gross Power (kW)	Total Gen Output (kW)	Net Plant Output (kW)	Dual Flash Output (kW)
Surprise Val.	375	663.0	485.0	51.5	12.41	2.06	.9112	.9270	.8579	14,285	46,623	55,917	48,221	49,998
Desert Peak	425	537.0	303.7	90.0	23.00	1.96	.8885	.9065	.8590	12,265	43,437	50,495	47,560	47,946
Dixie Valley	450	541.2	292.3	100.0	21.00	2.18	.8946	.9159	.8607	14,134	40,869	50,331	47,521	47,902
Glass Mtn.	510	547.3	223.0	151.0	28.00	2.09	.8833	.9099	.8633	16,051	42,488	54,148	51,347	51,339
Salton Sea	570	559.2	128.4	151.0	28.00	2.03	.8825	.9128	.8652	16,399	38,481	51,079	48,297	47,833

Table 7-3
Dual Flash/Steam Reheater Process Parameters and Specific Capital Cost

Site	Sat Liq Flow Rate (M lb/hr)	Sat Liq Temp (F)	Turb Exh Flow (M lb/hr)	Turb Exh Press (psia)	Turb Exh Quality	Rhtr Press Drop (psia)	Rhtr Duty (MM Btu/hr)	Rhtr LMTD (F)	Rhtr Area (ft sq)	Installed Cost (k $US)	Spec Capital Cost ($/kW)	Dual Flash Cost ($/kW)
Surprise Val.	6,641	285.5	663.0	15.41	0.949	2	53.72	40.30	33,325	716.6	2,325	2,242
Desert Peak	3,949	322.0	537.0	26.00	0.944	2	48.84	42.84	28,500	612.9	1,511	1,491
Dixie Valley	3,408	329.4	541.0	24.00	0.936	2	55.59	47.19	29,450	633.3	1,246	1,225
Glass Mtn.	2,415	360.1	547.0	31.00	0.925	2	64.79	51.60	31,372	674.7	1,514	1,504
Salton Sea	2,011	360.1	559.0	31.00	0.925	2	66.68	50.30	33,140	712.7	1,069	1,057

Sub-atmospheric Flash

Introduction

For the analysis of the baseline dual flash technology, the assumption is made that the low pressure flash pressure may not be below atmospheric pressure, in order to prevent any possible problems with air leakage into the system. This section analyzes the effect of relaxing this restriction: the resultant concept is called sub-atmospheric flash technology. It should be noted that the sub-atmospheric flash technology is only applicable to resources that have temperature below 400°F since the optimal low pressure flash pressure for resources with temperatures above 400°F is greater than one atmosphere.

Cycle Process Flow

The process design of a sub-atmospheric flash cycle is essentially the same as a conventional dual flash cycle. The reader is referred to the dual flash section of this report for the process description.

Performance Analysis

Description. Refer to the dual flash section for details on performance analysis of sub-atmospheric flash cycles.

Assumptions. The same assumptions that were used for commercial dual flash cycles apply here.

Cost Analysis

Description. The cost analysis methodology used in analyzing conventional dual flash plants was also used to analyze sub-atmospheric flash plants. In addition, the impact of sub-atmospheric turbine inlet pressures on turbine cost and performance was determined in consultation with Mr. Yuri Esaki, a co-investigator.

Results

Performance. Sub-atmospheric flash cycles were analyzed for three sites, Surprise Valley, Vale, and Raft River. Thermo Hot Springs with a resource temperature of 265°F was omitted because flash technology is unattractive at that low a resource temperature.

Table 7-4 presents a comparison of sub-atmospheric flash cycles with dual flash cycles. Figure 7-6 plots the specific output for both conventional dual flash and sub-atmospheric flash power plants. It can be seen from the figure that, in general, specific output is higher for sub-atmospheric flash cycles compared to conventional dual flash cycles but the increase in specific output is not dramatic.

It appears that the specific output is fairly constant over a range of flash pressures.

Table 7-4
Sub-atmospheric Flash Cycles: Specific Capital Cost and Output

	Specific Sub-atmospheric Flash	Output, kWh/1000# Dual Flash	Specific Subatmospheric Flash	Capital Cost $/kWh Dual Flash
Raft River	3.13	3.06	3,108	3,161
Vale	4.70	4.64	2,631	2,634
Surprise Valley	6.74	6.69	2,239	2,242

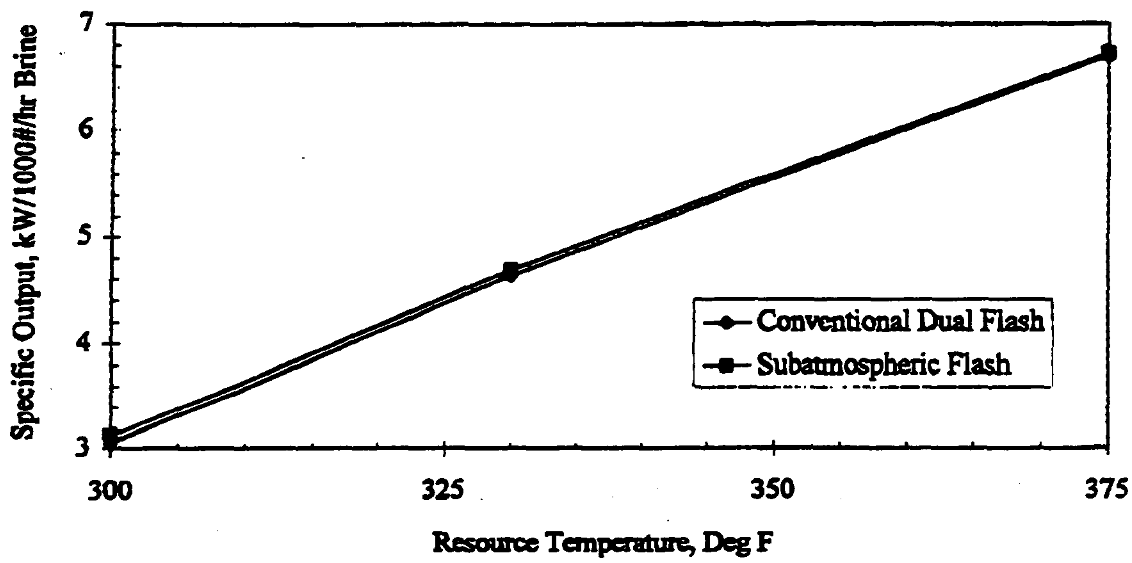

Figure 7-6
Sub-atmospheric Flash Cycle: Specific Output Comparison

Economics. Figure 7-7 compares the specific capital cost of the sub-atmospheric flash cycle with that of the conventional dual flash cycle. For Surprise Valley and Vale, the sub-atmospheric flash cycle is only slightly more economical than the conventional dual flash cycle. This is due to the fact that the brine utilization is only slightly better for these cases. For the case of Raft River, the sub-atmospheric flash cycle is about $80 per kilowatt cheaper.

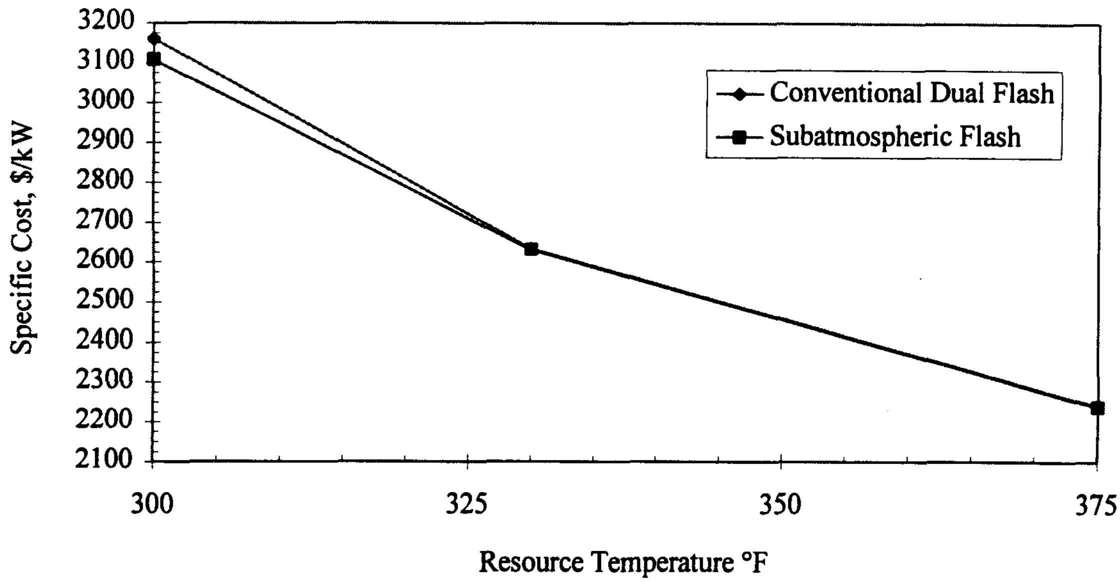

Figure 7-7
Sub-atmospheric Flash Cycle: Specific Cost Comparison

Although relaxing the restriction on the low pressure flash pressure leads to improvement over conventional dual flash technology, it does not make flash technology superior to binary technology at low resource temperatures. Binary technology still has better brine utilization and a lower capital cost per kilowatt.

It appears that sub-atmospheric inlet pressures do not affect turbine cost per se. However, lowering flash pressures below atmospheric increases the volumetric flow rate of steam because both the mass flow rate and specific volume increase. In turn, the higher volumetric flow rate increases exhaust losses. Consequently, an economic optimum exists between the added cost of using an extra turbine and its potential to improve power production by lowering exhaust losses. It was found that for the cases at Surprise Valley and Vale, two 27 inch last stage blade length dual flow turbines are still the best choice. This is the same as for the baseline dual flash cases. However, for Raft River, the two and three turbine plants were found to have nearly the same specific plant cost. Preliminary investigations on the Thermo Hot Springs case indicated that optimum turbine configurations required four or five turbines. Therefore, this case was not pursued any further.

Dual Flash/Hot Water Turbine

Introduction

The overall efficiency of flash plants can be improved by using hot water turbines to recover energy from the brine leaving the flash separators. A hot water turbine can be installed at the outlet of the flash separator in a single flash plant or between the high and low pressure separators in a dual flash plant. Although hot water turbines can be either impulse type or reaction type, the latter type is more suited to recovering energy from a hot water stream (Nishioka and Kato, 1994).

A full scale model of an axial flow reaction type hot water turbine has been tested successfully at a geothermal field in Japan (Nishioka and Kato, 1994). These authors report that after 1000 hours of operation the turbine parts showed no signs of erosion or corrosion. Moreover, deposits and scaling were not observed inside the nozzles and moving blades. The authors reported an internal efficiency of 38% and asserted that full scale models with an efficiency exceeding 40% could be designed.

Cycle Process Flow

Figure 7-8 contains a process flow diagram for the dual flash/hot water turbine cycle. The dual flash process is identical to that investigated in Section 5 of the report with the addition of the hot water turbine. The hot water turbine is employed between the high and low pressure separators producing added gross plant power.

Performance Analysis

This technology was applied at just three of the NGGPP sites: Raft River, Dixie Valley, and Glass Mountain. These three sites were chosen to represent a cold, medium, and hot resource, respectively, since resource temperatures determine the difference between the high and low flash pressures. Thus, with these three resources it is possible to examine the effect of the increasing hot water turbine pressure drop on the thermodynamics and economics of the power plants.

Description. Hot water turbine technology was analyzed using a slightly modified form of the Holt models used for analyzing dual flash cycles. In order to model the performance of the hot water turbine, a dual flash cycle was first simulated assuming the overall hot water turbine efficiency to be 40%. Using the liquid flow rate and the inlet and outlet pressures from these preliminary calculations, the hot water turbine was then designed by Fuji Electric Co. and the overall hot water turbine efficiency calculated. Overall efficiency calculated by Fuji was input into the dual flash model and the final net plant output computed.

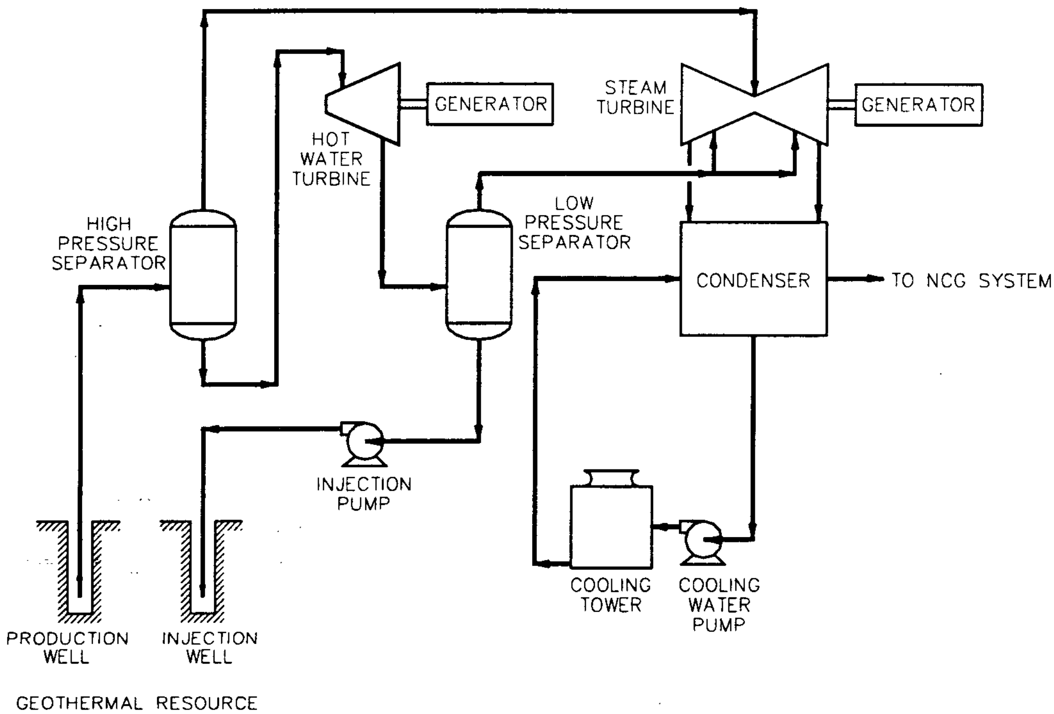

Figure 7-8
Dual Flash/Hot Water Turbine: Process Flow Diagram

Assumptions. The assumptions for the dual flash process are consistent with the conventional dual flash analysis (see Section 5). A 93% mechanical efficiency was assumed for the hot water turbine to account for generator, gear and bearing losses.

Cost Analysis

Description. Along with the turbine design, Fuji also provided detailed cost estimates. The cost of the hot water turbine is added to the dual flash cost model to calculate the entire dual flash/hot water turbine plant cost.

Assumptions. The installed cost multiplier for the hot water turbine is 1.8 because it was assumed that installation costs for a hot water turbine would be similar to those for an RST. Other assumptions are the same as those used to evaluate dual flash technology.

Results

Performance. Table 7-5 is a summary of each of the dual flash/hot water turbine cases. Several items are noteworthy. First, the initial estimate of forty percent efficiency for the hot water turbine was close to the final calculated values. Table 7-5 shows that the hot water turbine appears to perform more efficiently at Dixie Valley than at Glass Mountain. This is due to the fact that a standard turbine was used for all the sites. Ideally, hot water turbines should be tailor-made for each site to maximize performance. The hot water inlet pressure, a site-specific quantity, is used to optimize important turbine parameters including the ratio of moving blade throat area to nozzle throat area, nozzle outlet angle and profiles of nozzles and moving blades. However, optimizing the design of hot water turbines for these specific cases was deemed to be outside the scope of this study.

Table 7-5
Dual Flash/Hot Water Turbine Case Summaries

	Glass Mountain	Dixie Valley	Raft River
High Pressure Flash, psia	153.20	102.20	30.40
Low Pressure Flash, psia	18.00	15.00	13.45
Condenser Pressure, " Hg	2.16	2.24	1.78
Brine Flow Rate, lb/hr	3,000,000	4,000,000	15,391,851
High Pressure Steam Flow Rate, lb/hr	579,721	584,211	734,371
Low Pressure Steam Flow Rate, lb/hr	340,159	402,132	653,729
Hot Water Turbine Efficiency	0.41	0.46	0.38
Hot Water Turbine Power, kW	3,540	4,060	2,110
Steam Turbine Power, kW	53,230	49,280	60,224
Plant Net Power, kW	53,865	50,953	48,685

It was found that specific output can be maximized by lowering the low pressure flash pressure below the optimum found in the dual flash cases because allowing the hot water turbine to produce power over a larger pressure drop is thermodynamically favorable. At some point, however, the flash pressures become too low reducing the performance of the steam turbine by an amount more than the power gain from the hot water turbine.

Figure 7-9 compares the specific output of dual flash technology with and without the hot water turbine. As expected, the hot water turbine improves the specific output, and the improvement in specific output is greater for the hotter resources.

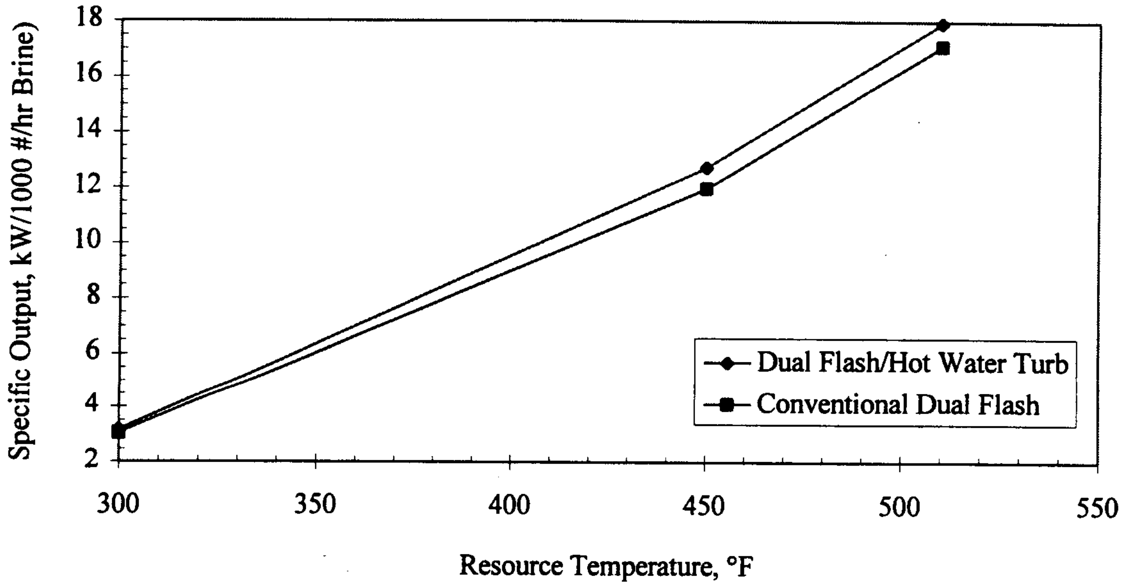

Figure 7-9
Brine Utilization Comparison

Economics. Major equipment and total plant costs for the dual flash/hot water turbine plant are summarized in Table 7-6. The installed cost of the hot water turbine-generator system divided by its gross power production for Glass Mountain and Dixie Valley is roughly \$1530/kW, and \$8000/kW for Raft River. These turbine-generator costs alone are higher than the specific capital cost for conventional dual flash technology. In addition, to some extent the hot water turbine produces power at the expense of the steam turbine, and thus the effect of using hot water turbines on specific capital cost is even greater. Figure 7-10 compares the specific capital cost of dual flash technology with and without a hot water turbine. As indicated earlier, specific capital cost of a dual flash plant with a hot water turbine is higher than that of a conventional dual flash plant.

Table 7-6
Dual Flash/Hot Water Turbine Plant Capital Costs

CASE	Raft River 300°F	Dixie Valley 450°F	Glass Mountain 510°F
Summary			
Plant Equipment Cost	30,796,903	18,251,466	19,012,863
Installed Plant Cost	77,916,000	46,176,000	48,103,000
Gath & Injec Equip Cost	3,306,695	25,000	25,000
Installed Gath & Injec Cost	6,977,000	53,000	53,000
Hot Water Turbine Cost	17,496,000	6,300,000	5,400,000
Gath & Injec Piping Cost	1,402,000	1,112,000	1,754,000
Well Cost	57,750,000	11,571,429	27,500,000
Total Plant Cost	161,541,000	65,212,429	82,810,000
Specific Plant Cost ($/kW)	3,318	1,280	1,537
Net Plant Output (MW)	48.7	50.9	53.9

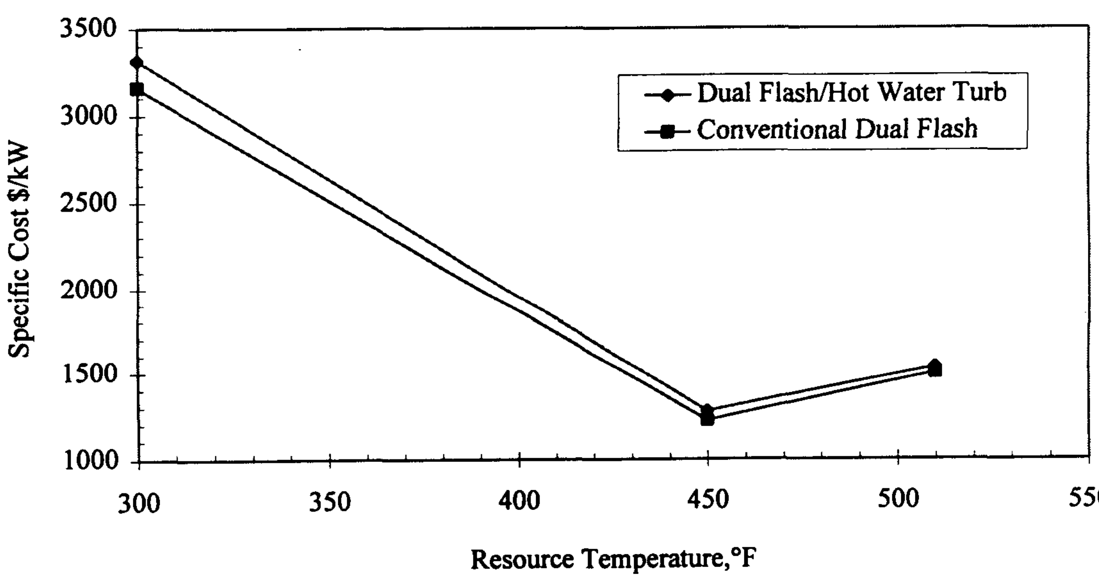

Figure 7-10
Specific Plant Cost Comparison

8

HYBRID CYCLES

Flash/Binary Bottoming Cycles

Introduction

The binary bottoming cycle is a combination of flash and binary technology. Flash technology is generally more cost effective for hot resources, while binary technology is more cost effective for cold resources: the hybrid plant attempts to use each type of technology in the temperature range where it is most cost effective.

Cycle Process Flow

Figures 8-1 and 8-2 show two simplified process flow diagrams for binary bottoming cycles. The first is a dual flash/binary bottoming hybrid plant. The single flash, binary bottoming hybrid plant process flow diagram is shown in Figure 8-2. The dual flash portion of the hybrid plant is identical to the dual flash cycle, discussed in Section 5, except that the brine from the low pressure separator is used to heat the hydrocarbon working fluid in the binary cycle before being injected into the reservoir. The binary cycle is identical to the air-cooled binary cycle discussed in Section 5.

The single flash/binary hybrid process is the same as the dual flash/binary hybrid except that the flash plant does not have a low pressure separator, and has a single entry turbine. In the single flash cycle the brine from the high pressure separator is used to heat the binary cycle.

Performance Analysis

Description. Flash/binary hybrid cycles were analyzed using the methodologies used to analyze dual flash and binary cycles. The temperature and enthalpy of the brine from the separator were used as an input for the binary plant performance model. Binary cycle net power was input into a modified dual flash model which was used to calculate the overall hybrid plant net power.

Since the flash pressure determines the amount of power that will be produced by each part of the hybrid plant, a case was run at the dual flash optimum flash pressures, and then several cases were run raising the low pressure flash pressure incrementally until there was only a single flash.

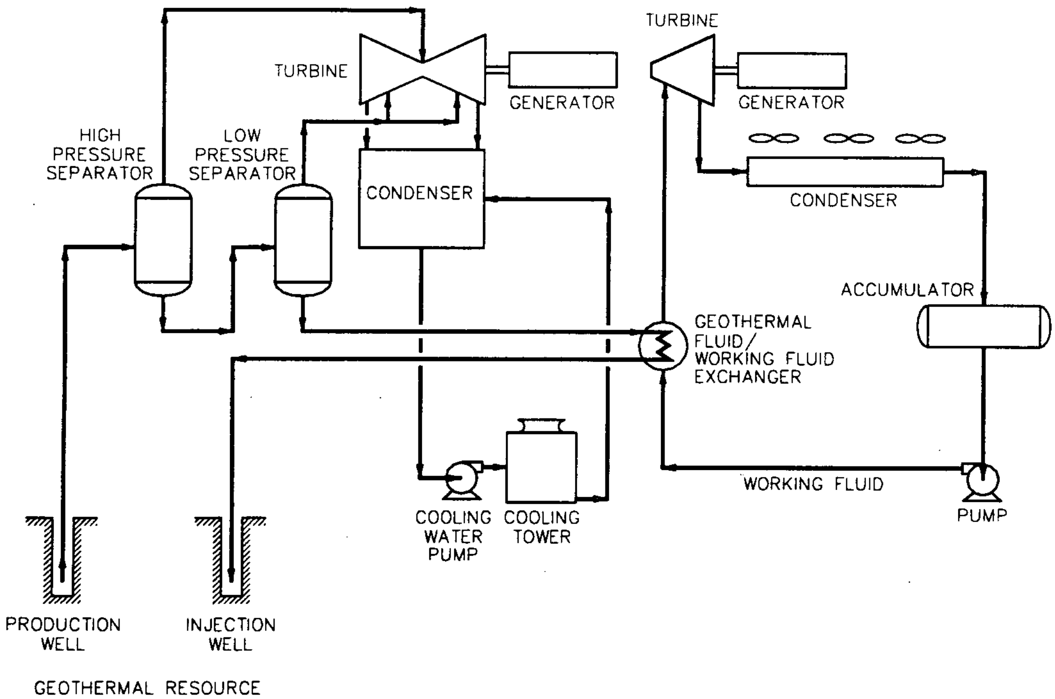

Figure 8-1
Dual Flash/Binary Bottoming Cycle: Process Flow Diagram

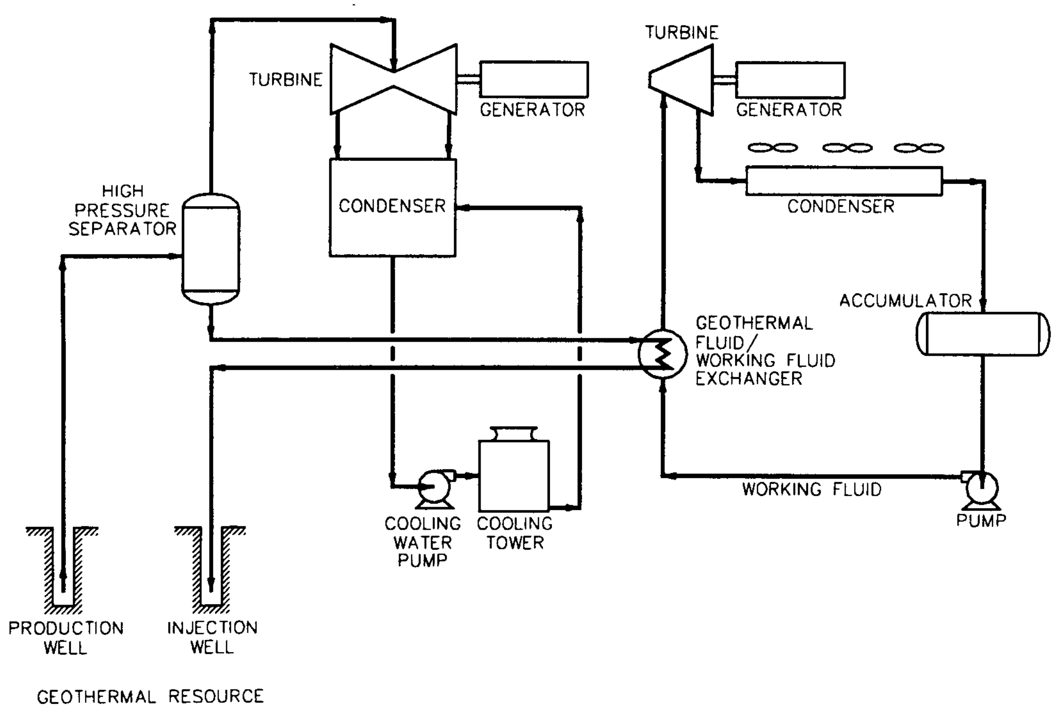

Figure 8-2
Single Flash/Binary Bottoming Cycle: Process Flow Diagram

As the low pressure flash pressure increases, the amount of low pressure steam to the steam turbine decreases, and more and hotter brine is sent to the binary plant. The low pressure flash pressure which gives the maximum plant output was determined for each case.

Assumptions. Assumptions used in the analysis of binary and dual flash cycles are applicable to the flash/binary hybrid cycles.

Cost Analysis

Assumptions. Economic analysis of the binary bottoming hybrid plants was performed using the same methods and assumptions as those used to calculate the cost of flash plants and binary plants individually. Offsite costs such as well costs, and gathering and injection system costs were included in the flash portion of the plant cost.

Results

Performance. The results of the thermodynamic analysis are shown on Table 8-1. Each row represents one case. The second column shows the low pressure flash pressure as it is raised incrementally from the dual flash economic optimum up to a single flash. The third column shows the brine utilization for each case. The brine utilization in all cases is better than that for dual flash alone. Furthermore, it is thermodynamically favorable to raise the low pressure flash pressure above the dual flash optimum to divert more and hotter brine to the binary part of the plant. However, as more and hotter brine is diverted to the binary part of the plant, less low pressure steam is available for the flash part of the plant, and there is a point of diminishing return where total net power production levels off. In Table 8-1, the optimum brine utilization is in bold face.

For the Glass Mountain site, the thermodynamic optimum plant low pressure flash is 100 psia. For the Desert Peak site, the optimum is about 50 psia. For the more cold resources at Surprise Valley, Vale, and Raft River, it is best to eliminate the second flash altogether and use a single flash hybrid plant.

Economics. The results of the economic analysis are presented along side the results of the thermodynamic analysis on Table 8-1. The fourth column gives the specific capital cost of the hybrid plant. The fifth and sixth columns give the specific capital cost of the flash and binary parts of the plant respectively.

The first cases for each site have the same flash pressures as the economic optimum dual flash cases. In each of these cases, the specific cost of the dual flash part of the plant is slightly more than the conventional dual flash specific capital cost (refer to Table 5-7). This is due to the fact that these flash portions of the hybrid plants are smaller, and the economy of scale makes the full size dual flash plant less expensive.

Table 8-1
Summary of Flash/Binary Bottoming Cycle Cases

Site		Low Flash Press (Psia)	Brine Utilization (kW-hr / 1K lb brine)	Total Cost ($/kW)	Flash Cost ($/kW)	Binary Cost ($/kW)	Flash Output (MW)	Binary Output (MW)
Glass Mountain		26.0	18.78	1648	1565	2568	46.54	4.17
		35.0	19.42	1642	1562	2229	46.17	6.27
		50.0	19.44	**1635**	1586	1931	45.07	7.40
		70.0	19.71	1644	1632	1697	43.15	10.05
		100.0	**20.03**	1690	1719	1608	40.05	14.03
	(1)	151.0	19.68	1786	1857	1651	34.83	18.32
Desert Peak		21.5	11.77	1565	1497	2233	48.02	4.93
		30.0	12.15	**1544**	1501	1824	47.35	7.37
		50.0	**12.47**	1595	1576	1660	43.86	12.26
	(1)	90.0	12.32	1721	1782	1616	34.97	20.46
Surprise Valley		12.5	7.15	2191	2055	3074	42.11	6.48
		25.0	7.86	**2129**	2063	2335	40.59	12.85
	(1)	51.5	**7.93**	2179	2493	1789	29.86	24.02
Vale, Oregon		12.5	4.83	2674	2624	3037	36.94	9.78
		20.0	5.17	**2669**	2708	2582	34.85	15.65
	(1)	35.4	**5.21**	2759	3432	2123	24.53	26.01
Raft River		12.5	3.80	3341	3498	2963	33.90	14.05
		20.0	3.88	**3068**	3665	2360	26.50	22.37
	(1)	28.2	**3.91**	3167	4862	2181	18.13	31.17

(1) Single Flash/Binary Bottoming Cycle

Looking at the first Glass Mountain hybrid case in Table 8-1, it can be seen that the binary part of the plant is operating with a 242°F (saturation temperature of steam at 26 psia) resource. The cost of binary power production for such a cold resource is very high as seen in the air-cooled binary plant analysis in Section 5. As the flash pressure is raised, the cost of the flash part of the plant rises because of the diminishing scale as well as the fact that the flash pressures diverge from the optimum, but the cost of the binary plant decreases because it receives a hotter resource. Thus, an economic optimum exists for each site.

Table 8-2 presents a comparison of the flash/binary hybrid cycles with dual flash and binary cycles. At Surprise Valley, Vale and Raft River, the specific capital cost of the hybrid cycle is higher and its specific output lower than the air-cooled binary cycle. Thus, at these three sites the baseline technology would be preferred over the hybrid. At Glass Mountain and Desert Peak, the hybrid cycles have a higher specific capital cost but also a higher specific output compared to dual flash cycles. In fact, at these two resources the high specific output of the dual flash/binary hybrid cycle does make it the technology of choice as will be seen in Section 10.

Table 8-2
Flash/Binary Bottoming Comparison Summary

Site	Specific Output, kWh/1000 #			Specific Capital Cost, $/kW		
	Flash/ Binary	Dual Flash	Binary	Flash/ Binary	Dual Flash	Binary
Glass Mountain	19.44	17.11	13.06	1,635	1,504	2,070
Desert Peak	12.15	10.65	-	1,544	1,491	-
Surprise Valley	7.86	6.69	8.49	2,129	2,242	2,115
Vale	5.17	4.64	5.99	2,669	2,634	2,356
Raft River	3.88	3.06	4.04	3,068	3,161	2,945

Dual Flash/Back-Pressure-Turbine Binary

Introduction

While the binary bottoming cycle is a conventional flash plant with a conventional binary plant added down stream, the back-pressure-turbine cycle combines flash and binary technology with major modifications, as discussed below, to each of the processes.

Cycle Process Flow

Figure 8-3 shows a simplified process flow diagram for this cycle. The major modification to the dual flash part of the plant is the elimination of the cooling water system. A steam-hydrocarbon heat exchanger functions as the steam condenser for the dual flash process. The hydrocarbon side of this exchanger is used to heat the working fluid in the binary cycle. A compressor is used to pressurize the noncondensable gases (NCGs) in the steam so that they can be injected into the reservoir.

The assumption that NCGs can be successfully disposed of by injecting them back into the reservoir at any geothermal site is at best questionable because it is almost never used in the industry. Injecting NCGs into the reservoir is environmentally desirable because it virtually eliminates harmful emissions. Moreover, NCGs can be injected using a compressor which is less expensive than a sulfur plant which would be required to treat noncondensable gases if they were not injected. For the same resource and capacity, a power plant based on NCG injection is about 2% lower in cost than a power plant design which includes a sulfur plant for treating NCGs. In spite of these significant advantages, NCG injection is rarely used in geothermal power plants. In part this is due to the potential of injected NCGs to flow back out of the reservoir through the production wells and thereby adversely affect brine production.

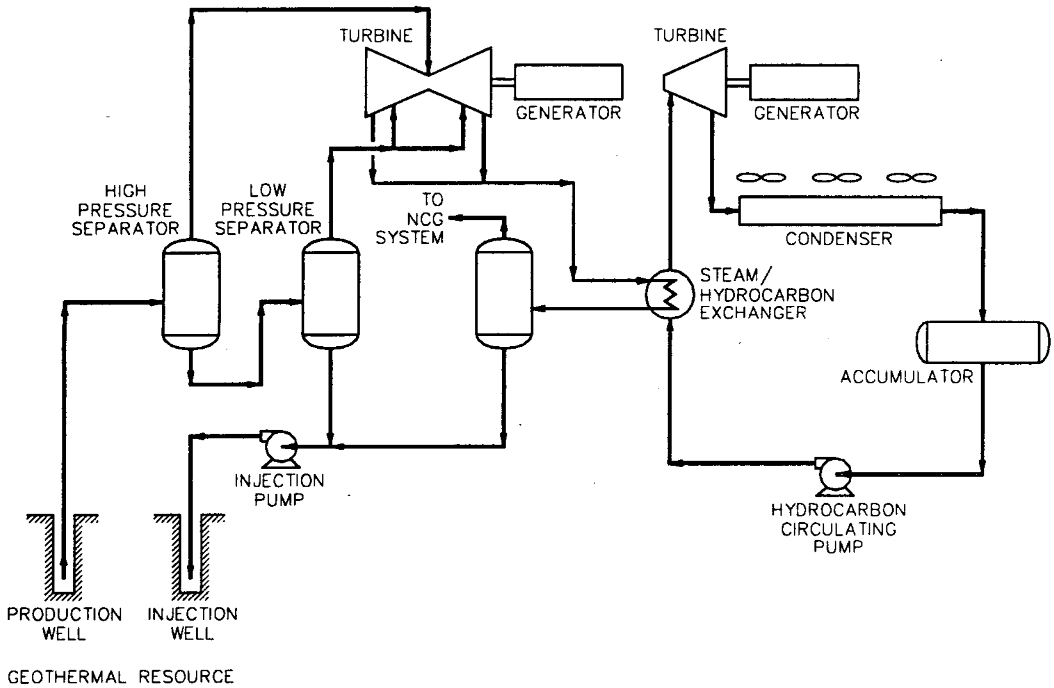

Figure 8-3
Dual Flash/Back-Pressure-Turbine Binary: Process Flow Diagram

Thus, clearly the industry has reservations about this technology. In this study the dual flash/back-pressure-turbine binary hybrid cycle is designed with NCG injection in order to make at least one flash cycle a closed cycle. It is presumed that closed flash cycles would be the lowest cost flash cycles and would serve as an important point of reference for the NGGPP. In sum, a dual flash/back-pressure-turbine binary hybrid cycle with NCG injection can perhaps be implemented only in the rare circumstance but it does serve as an ideal to which other technologies can be compared.

The components of the flash part of the plant upstream of the steam turbine are the same as those in a conventional dual flash plant. Likewise, except for the steam-hydrocarbon heat exchanger, the binary part of the plant is the same as an air-cooled binary cycle power plant.

Performance Analysis

Assumptions. In analyzing the dual flash/back-pressure-turbine binary cycles, the steam side pressure at the outlet of the condenser was fixed at one atmosphere, and a two pound pressure drop on the steam side of the condenser was assumed. Thus, the turbine back pressure was set at two pounds above atmospheric. For the resource temperatures of interest, this makes for roughly equal contributions to the total net power output from both parts of the hybrid plant when the cycles are optimized. Decreasing the back pressure leads to a greater contribution of the power output from the steam turbine. Along with this, however, comes the potential for air leakage into the process, and the requirement for a greater amount of power to compress the noncondensable gases for injection.

The binary turbine inlet temperature was fixed by setting the approach temperature on the condenser and specifying the air cooler approach temperature fixed the hydrocarbon state points. Since the back pressure of the steam turbine is nearly the same for all cases, varying only with the atmospheric pressure at the individual sites, the binary cycle state points are nearly the same for all the hybrid plant cases. The amount of steam condensed determines the condenser duty, the amount of working fluid circulation, and therefore the amount of power produced by the binary cycle.

Cost Analysis

Assumptions. The economic analysis methodologies used for flash and binary cycles were also used for analyzing dual flash/back-pressure-turbine binary hybrid power plants. The costs for the condenser, cooling water system, vacuum system and the sulfur plant were omitted from the dual flash cost model and the cost of a gas compressor was added. Offsite costs were included in the flash part of the plant. The only adjustment made to the binary plant cost model involved changing the overall heat transfer coefficient on the hydrocarbon heater. Heat exchanger costs were assumed to be $7.57 per square foot of surface area, the same as those for the brine-to-isobutane heat exchanger in the binary cycle.

Results

Performance. The flash pressures in the flash part of the plant affect the power output of both parts of the hybrid plant. Figure 8-4 shows the power produced by each part of the plant and the total plant power as a function of low pressure flash pressure at Glass Mountain. The high pressure flash pressure is kept at a constant 151 psia. Since this is the wellhead pressure, there is no ability to raise it, and it is thermodynamically unfavorable to lower it. The plot shows that the optimum low pressure flash pressure is about 53 psia.

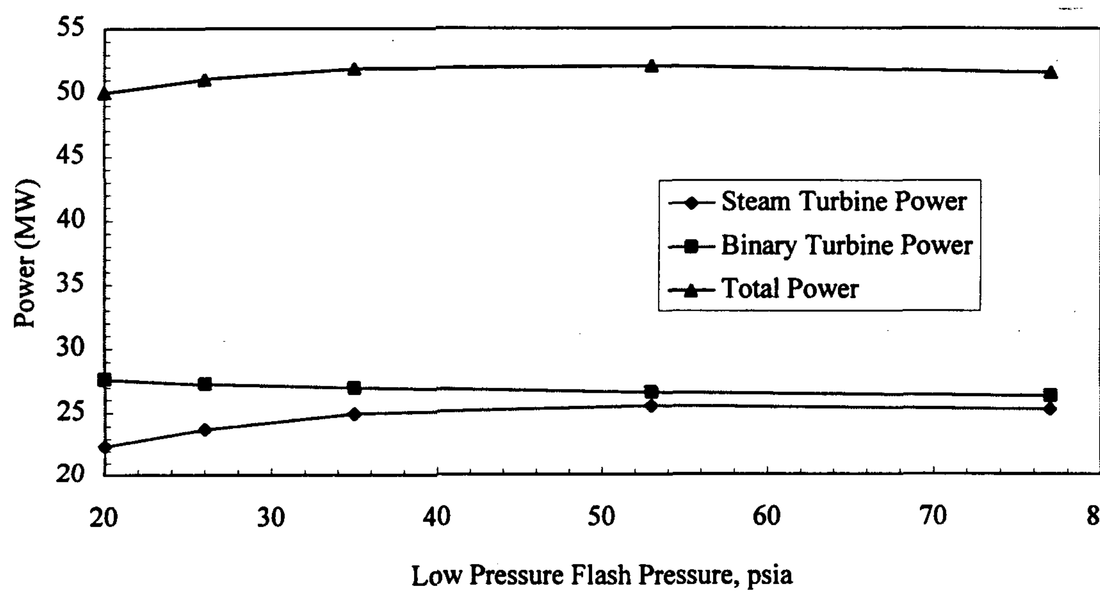

Figure 8-4
Dual Flash/Backpressure Turbine: Flash Pressure Optimization, Glass Mountain

The optimum flash pressure for the hybrid is higher than the 26 psia optimum for the conventional dual flash case because the condenser temperature in the back-pressure-turbine cycle is higher. As a result, less power is produced in the binary part of the hybrid cycle because increasing the flash pressure produces a smaller quantity of low pressure steam.

A performance summary for the optimum dual flash/back-pressure-turbine binary hybrids is presented in Table 8-3. The specific output for this hybrid cycle at Surprise Valley and Vale, the two colder resources, is higher than that for the conventional dual flash cycle but it is lower than that for binary cycles. Thus, at Surprise Valley and Vale the performance of a dual flash cycle can be improved by combining it with a backpressure turbine binary hybrid but it would remain significantly inferior to the performance of binary cycles at these resources.

For the hotter (than 375°F) resources, the performance of dual flash/ backpressure turbine binary cycle is comparable to that of dual flash cycles as can be seen from Table 8-2. The dual flash cycle has a higher specific output than the hybrid at Coso because the resource at Coso has a high NCG content. As a result, the hybrid cycle compressor parasitics associated with NCG injection are relatively high. For the other sites it appears that the advantage of heat recovery (in the binary part) afforded by the hybrid cycle is more or less offset by higher turbine backpressure and the irreversibilities introduced into the cycle by the heat recovery process.

Table 8-3
Dual Flash/Backpressure Turbine Binary Hybrid Performance Summary

Site	Specific Output kWh/1000#	Flash Spec. Output kWh	Binary Spec. Output kWh	Condenser Pressure psi	LP Flash P psia	Net Output kW
Glass Mountain	17.36	17.11	13.06	14.2	53.0	52,087
Coso	15.55	16.25	–	14.6	51.0	49,747
Desert Peak	10.41	10.65	–	14.4	41.0	46,862
Surprise Valley	7.42	6.69	8.49	14.4	39.5	55,437
Vale	4.96	4.64	5.99	14.6	31.0	61,724

Economics. The back-pressure-turbine cycle has several parameters which must be optimized. These are parameters that normally have to be optimized in either dual flash or binary plants when they are investigated separately. A plot of specific plant cost vs. low pressure flash pressure for Glass Mountain (Figure 8-5) shows that the thermodynamic optimum flash pressure, 53 psia, is also the economic optimum. This result with regard to the flash pressures was true for all of the back-pressure-turbine cycles investigated and the same result was found with the dual flash investigations as well.

For each case, an approach temperature on the condenser of 15°F is found to be the most cost effective. The amount of surface area, and therefore condenser cost, required to give a lower approach temperature is not justified by the amount of added power produced. For the air coolers, an approach temperature of 28°F is found to be the most cost effective. These temperatures were found to be optimal for all cases because the binary cycle heat source and rejection temperatures are approximately equal for all cases.

Table 8-4 gives the major equipment and total plant costs for the most economical plant at each site. The table also lists the specific capital cost for the hybrid plant along with the specific capital cost for the dual flash and binary

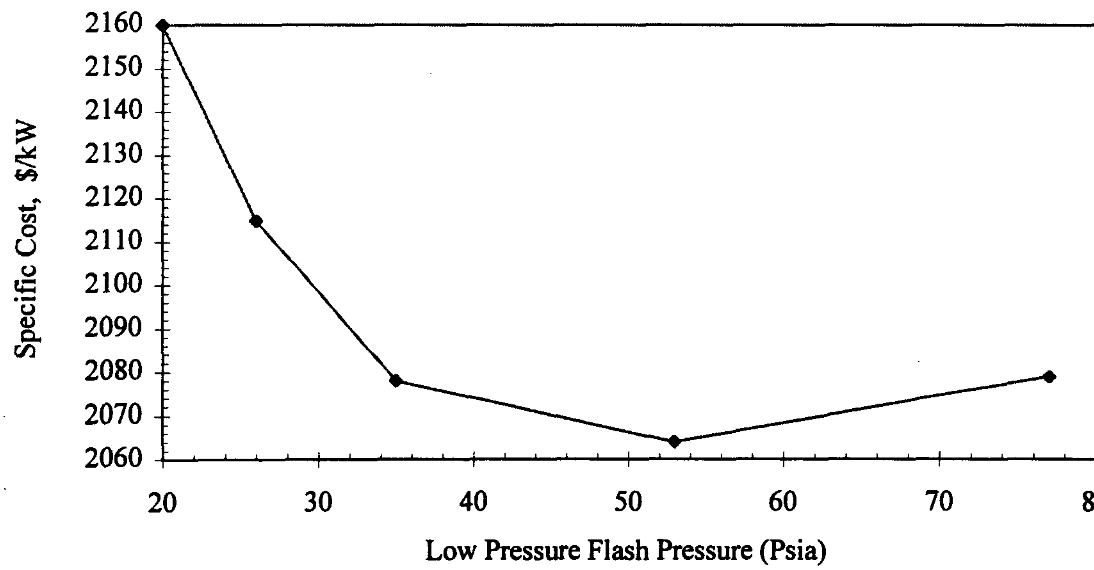

Figure 8-5
DFBPT: Specific Plant Cost v. LP Flash Pressure, Glass Mountain

plants. It is clear from the table that as the resource temperature declines, the power production and therefore the plant cost shifts from the flash part of the plant to the binary part of the plant.

Table 8-4 shows that the specific capital cost for the hybrid cycles is significantly higher than the binary cycles at Surprise Valley and Vale, and the flash cycles at the other sites. This result can be ascribed in large part to economies of scale and a combination of other factors. Compared to the large, single economical steam turbine used in the dual flash cycle, the hybrid cycle uses smaller, uneconomical turbines. For example, at Coso the turbine-generator cost for the hybrid plant would be almost $10 million higher than that for the dual flash plant. Further, hybrid cycles use air coolers to reject heat which are more expensive than the cooling water system and condensers that are used in dual flash plant. At Surprise Valley and Vale, the hybrid cycles suffer because the wellfield costs for the hybrid cycles are higher than those for the binary cycles since the specific output of binary cycles is 14-20% higher than that of the hybrid cycles.

Table 8-4
Dual Flash/Back-Pressure-Turbine Binary - Major Equipment and Total Plant Costs

CASE	Glass Mtn 510 °F	Coso 525 °F	Desert Pk 425 °F	Surprise Val. 375 °F	Vale 330 °F
Summary					
Plant Equip. Cost	10,820,603	13,386,915	9,671,645	11,874,219	12,580,554
Installed Flash Plant Equip Cost	27,376,000	33,869,000	24,469,000	30,042,000	31,829,000
Installed Gath & Injec Equip Cost	53,000	53,000	53,000	3,510,000	5,710,000
Installed Binary Plant Cost	50,740,312	48,829,196	51,014,294	76,564,344	100,071,439
Gath & Injec Piping Cost	1,733,000	1,540,000	1,486,000	1,084,000	1,248,000
Well Cost	27,500,000	22,000,000	23,765,060	31,000,000	45,264,706
Total Plant Cost	107,402,312	106,291,196	100,787,354	142,200,344	184,123,145
Specific Capital Cost	107,402,312	106,291,196	100,787,354	142,200,344	184,123,145
Hybrid, $/kW	2,062	2,137	2,151	2,565	2,983
Dual Flash, $/kW	1,504	1,588	1,491	2,242	2,634
Binary, $/kW	2,070	-	-	2,115	2,356

Dual Flash/Gas Turbine Hybrid

Introduction

Dual flash/gas turbine hybrid power plants combine geothermal power production with fossil fuel power production to increase the efficiency of each process over independent operation. What is otherwise waste heat from the gas turbine exhaust is transferred to the geothermal fluid to increase the enthalpy available.

Cycle Process Flow

Several types of dual flash/gas turbine hybrid plants were investigated, and process flow diagrams for the two best alternatives are shown on Figures 8-6 and 8-7. The dual flash portion of the plant is the same as described previously with the addition of one heat exchanger in each case. For the resources which require pumping, those below 400°F, the gas turbine exhaust heats the liquid resource before it enters the high pressure separators (Figure 8-6). The free flowing resources, those above 400°F, employ a heat recovery steam generator (HRSG) which makes additional high pressure steam from the condensate (Figure 8-7).

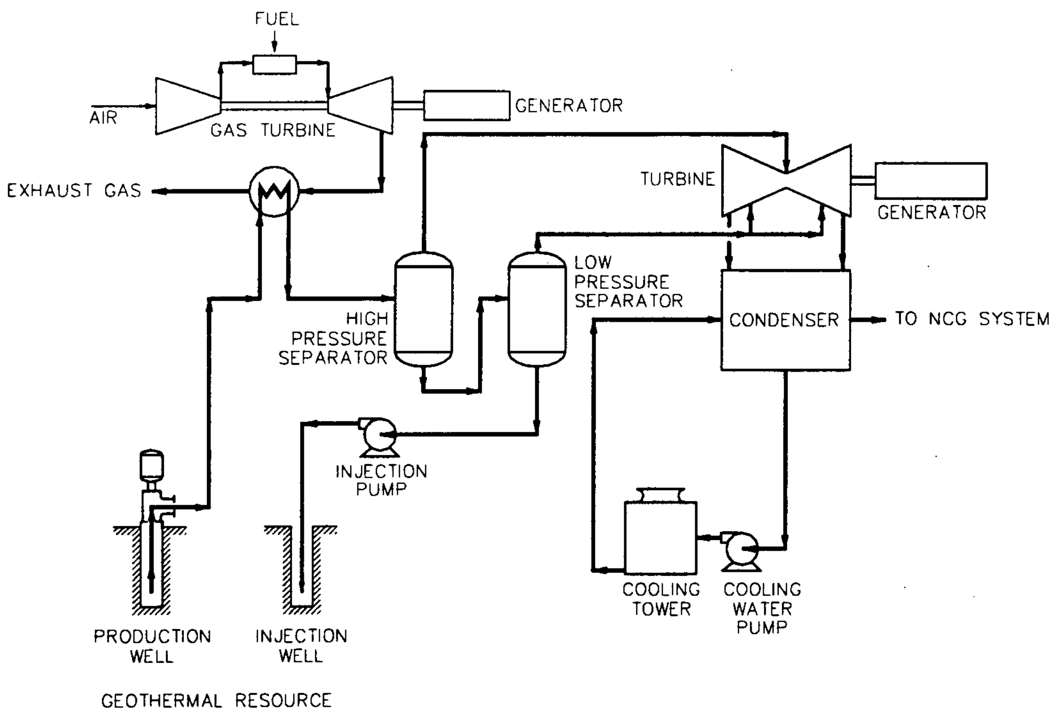

Figure 8-6
Dual Flash/Gas Turbine Hybrid I: Process Flow Diagram

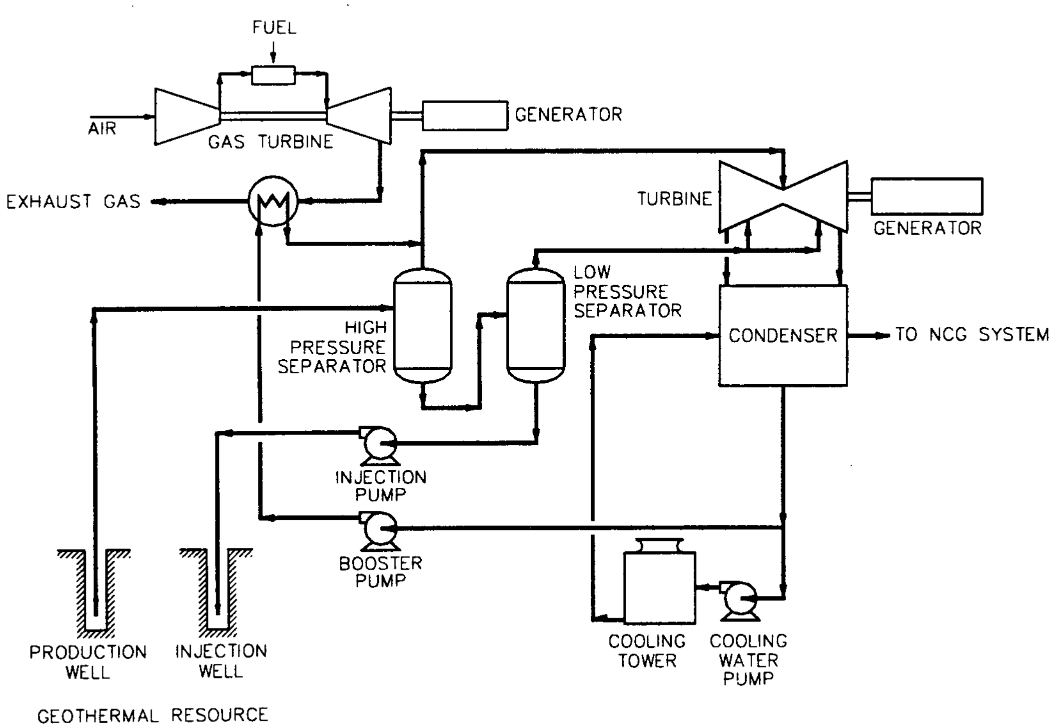

Figure 8-7
Dual Flash/Gas Turbine Hybrid II: Process Flow Diagram

Performance Analysis

Description. A General Electric (GE) LM2500 was used at each site because this unit produces roughly 20 MW, or forty percent, of the total net power for the hybrid plant. Since geothermal power is the subject of this study, it was decided that for all cycles the majority of the power should be produced by the geothermal part of the plant. The net power output, fuel consumption, exhaust gas enthalpy and exhaust gas flow rate for the LM2500 were obtained found from curves published by GE. These values are a function of atmospheric pressure and ambient weather conditions. Since in this study the wet bulb and dry bulb temperatures are assumed to be constant, the power output of the gas turbine and the enthalpy available in the exhaust gas are the same for each site.

The remaining task is to utilize the exhaust gas heat to augment the geothermal portion of the cycle. The geothermal cycle is optimized to minimize the brine required to generate a nominal 50 MW (net) for the combined plant.

For pumped resources, liquid geothermal fluid is pumped to that part of the plant where the gas turbine is located. Heat from the gas turbine exhaust is used to raise the temperature of the geothermal fluid, increasing its effective enthalpy. The increased enthalpy brine is then fed to a dual flash plant.

The high pressure separators for the free flowing resources are located at the production well pads. Therefore, it is not practical to use the exhaust gas from the gas turbine, which is located in the main plant, to supply heat to the fluid entering the high pressure separators. To minimize the technical risk associated with adding this heat, cycles involving superheating geothermal steam or vaporizing geothermal brine were eliminated. The superheated geothermal steam would be highly corrosive and an exhaust gas-to-superheated steam exchanger would have poor heat transfer requiring excessive surface area. Vaporizing geothermal brine could cause excessive fouling as the dissolved solids are deposited on the heat exchanger.

Three options were considered for recovering gas turbine exhaust heat for the free flowing cases. Option one involves heating the high pressure separator liquid effluent prior to being flashed in the low pressure separator. Option two involves heating a portion of the turbine condensate to generate high pressure steam. Option three, involves evaporating a portion of the condensate to high pressure steam. These three options were evaluated using the Desert Peak resource with a brine flow rate of 4,250,000 lb/hr. Table 8-5 summarizes the results of this evaluation. As can be seen from the table, option three produces the most power. It is also the simplest alternative, employing a standard HRSG as the heat exchanger. Therefore, option three was used for the free flowing cases.

Table 8-5
Dual Flash/Gas Turbine Hybrid - Evaluation of Cycle Alternatives

Option		1	2	3
Description	No Gas Turbine	Heat LP Brine	Heat Turbine Condensate	Evaporate Condensate
Gross Power (kW)	51,092	54,138	53,057	54,809
Net Power (kW)	48,047	50,450	49,461	51,349
Plant Net (kW)	48,047	69,162	68,353	70,061
LP Flash (psia)	22.5	27	22.5	22.5
Cond Press (in Hg)	1.86	1.9	1.89	1.96
CT Circ. (gpm)	56,000	67,000	66,000	66,500
Exh Inlet (F)		980	980	980
Exh Outlet (F)		274	698	352
Geoth Inlet (F)		244	234	100
Geoth Outlet (F)		344	320	320
Hx Duty (MMBtu/h)		87	36	78
Hx LMTD (F)		198	593	305
Pump Power (kW)		229	3	8

Assumptions.

- The assumptions for hybrid analysis are consistent with those for conventional dual flash analysis for similar process components.
- A minimum exhaust gas exit temperature of 260°F was set to limit condensation.
- There was a 30°F limit on the approach temperature for the HRSG. This is approximately an economic optimum.

Cost Analysis

Description. The capital cost of the LM2500 was assumed to be fixed at $8,600,000 which is based on a quote from G.E. The costs of the HRSG and the brine heater were found as a function of surface area as are all heat exchangers in the study. The optimum plant is defined as one which produces the required amount of power for the lowest capital cost (50 MW net minus the gas turbine power).

Assumptions.

- The cost assumptions were consistent with those made in the dual flash economic analysis for similar equipment.

- The installed cost multiplier for the LM2500, the HRSG, and the brine heater is 1.8 and was calculated using data published in the Gas Turbine World 1990 Handbook (Gas Turbine World, 1990).

Results

Performance. The geothermal part of the hybrid plants, with the exception of Thermo Hot Springs, can produce the 30 MW of power with a single steam turbine. Thermo Hot Springs still requires two turbines even though the steam flow is significantly reduced from the conventional 50 MW dual flash cycle.

The dual flash portion of the hybrid plant uses roughly half of the flow of the full scale plant yet produce over 60 percent of the net power. This is generally true for all of the resources. The added heat from the gas turbine exhaust produces roughly a twenty percent increase in brine utilization. Figure 8-8 shows the specific output comparison.

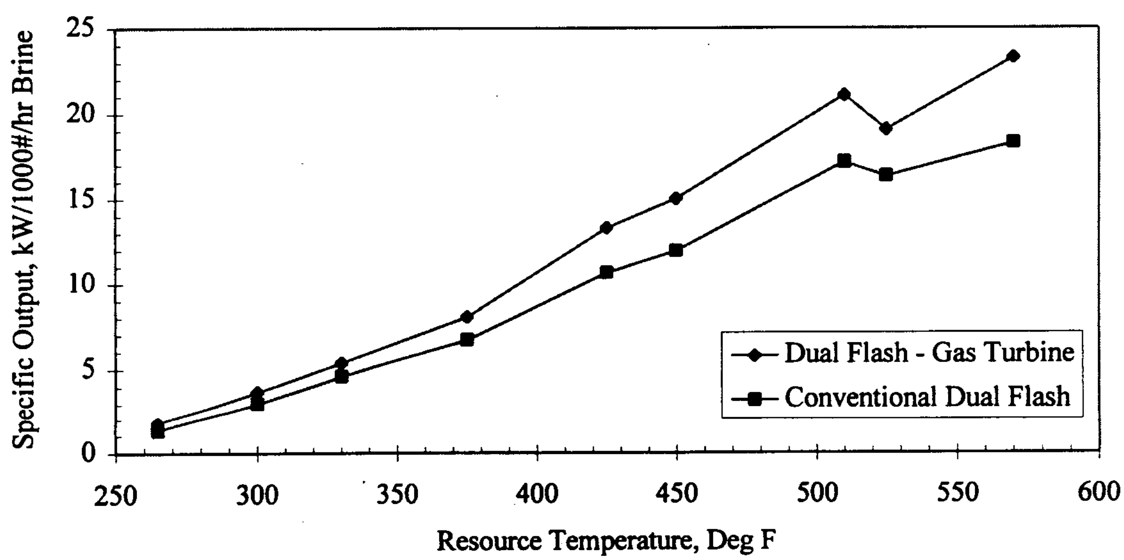

Figure 8-8
Dual Flash/Gas Turbine Hybrid - Specific Output v Resource

Economics. The specific capital cost of dual flash/gas turbine hybrids is not directly comparable to the all-geothermal technologies because although the capital costs are significantly lower, the busbar costs for gas turbine cycles, unlike geothermal cycles, include a significant fuel cost component. Therefore, a low capital cost does not necessarily translate into a low busbar cost, as is the case with the all-geothermal cycles. A comparison of the levelized busbar costs is presented in Section 10.

Binary/Gas Turbine Hybrid

Introduction

In this hybrid concept, exhaust gas from a natural gas fired combustion turbine is used to provide additional heat input to the binary cycle working fluid. The addition of high temperature exhaust heat increases the thermodynamic efficiency of the binary cycle operating on a relatively low temperature geothermal fluid.

Cycle Process Flow

As in the dual flash/gas turbine hybrid cycle, the gas turbine is a General Electric LM 2500. Figure 8-9 shows a process flow diagram for the binary/gas turbine hybrid with the binary and exhaust gas exchangers in series. In this cycle, cold isobutane is heated and partially vaporized in the geothermal fluid-to-isobutane exchanger and then heated to turbine inlet conditions in the exhaust gas-to-isobutane exchanger.

Figure 8-10 shows a cycle with a parallel heat exchanger configuration. In this cycle the cold isobutane is heated to its bubble point in a brine preheater. After leaving the brine preheater, the isobutane flow splits with some of the isobutane being vaporized in the exhaust gas-to-isobutane vaporizer and some in the geothermal fluid-to-isobutane vaporizer. The parallel heat exchanger configuration would be thermodynamically less favored for supercritical cycles because heat would be recovered at a lower temperature with this configuration. Consequently, cycles with parallel exchangers were not evaluated for Surprise Valley and Vale.

Figure 8-11 shows a temperature-enthalpy diagram for Thermo Hot Springs of the series heat exchanger configuration. Figure 8-12 shows a temperature-enthalpy diagram for Thermo Hot Springs of the parallel heat exchanger configuration. The diagrams show the temperatures of the various fluids on the vertical axis and the enthalpy, in Btu/lb, referenced to the working fluid on the horizontal axis.

Assumptions

For all cases, the combined power output of the binary and gas turbine portions of the plant was set at 50 MW. Cycle and economic analyses of the binary/gas turbine hybrid were performed for the low temperature resources where a binary cycle is favored over dual flash (Thermo Hot Springs, Raft River, Vale and Surprise Valley).

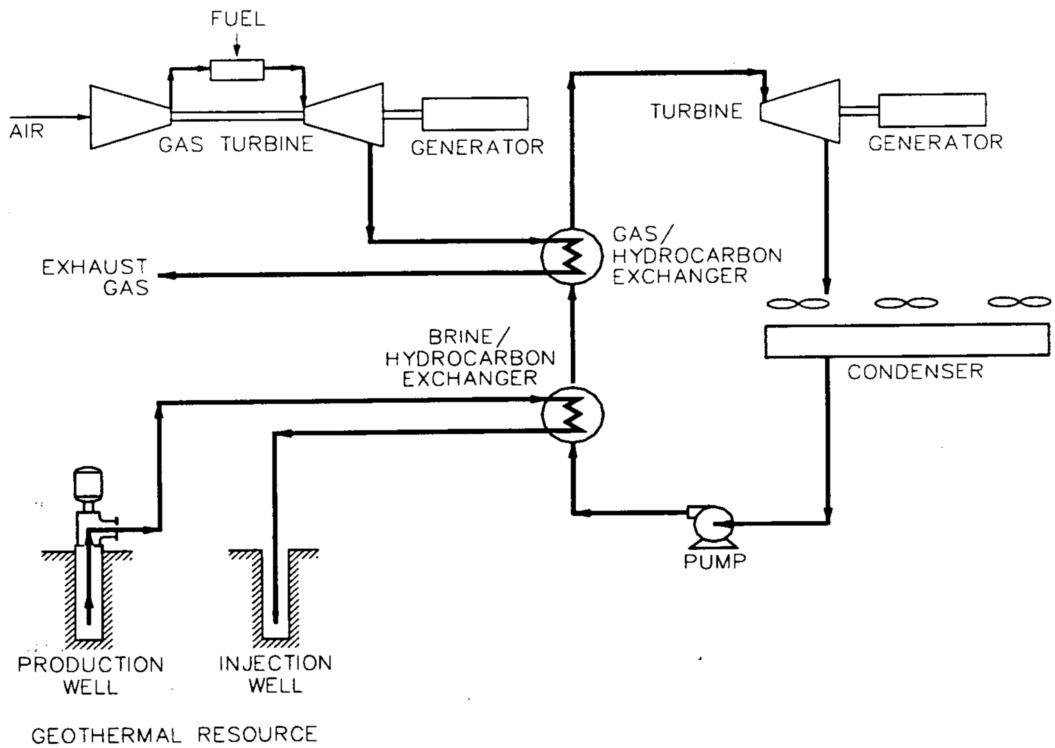

Figure 8-9
Gas Turbine - Binary Hybrid I: Process Flow Diagram

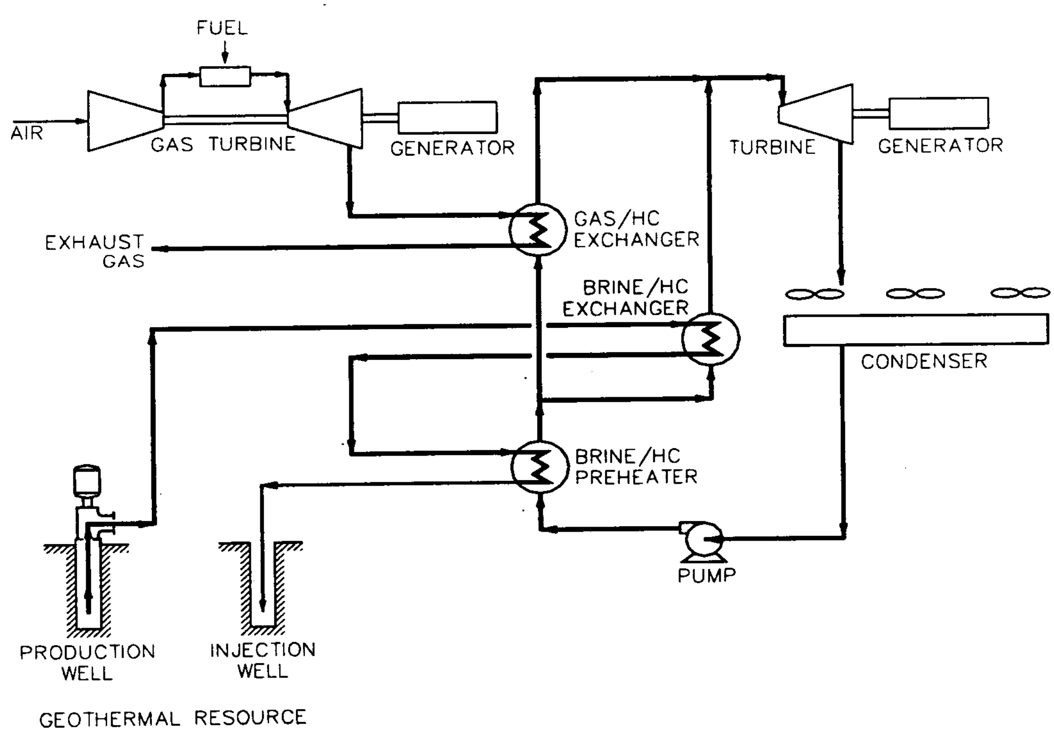

Figure 8-10
Gas Turbine - Binary Hybrid II: Process Flow Diagram

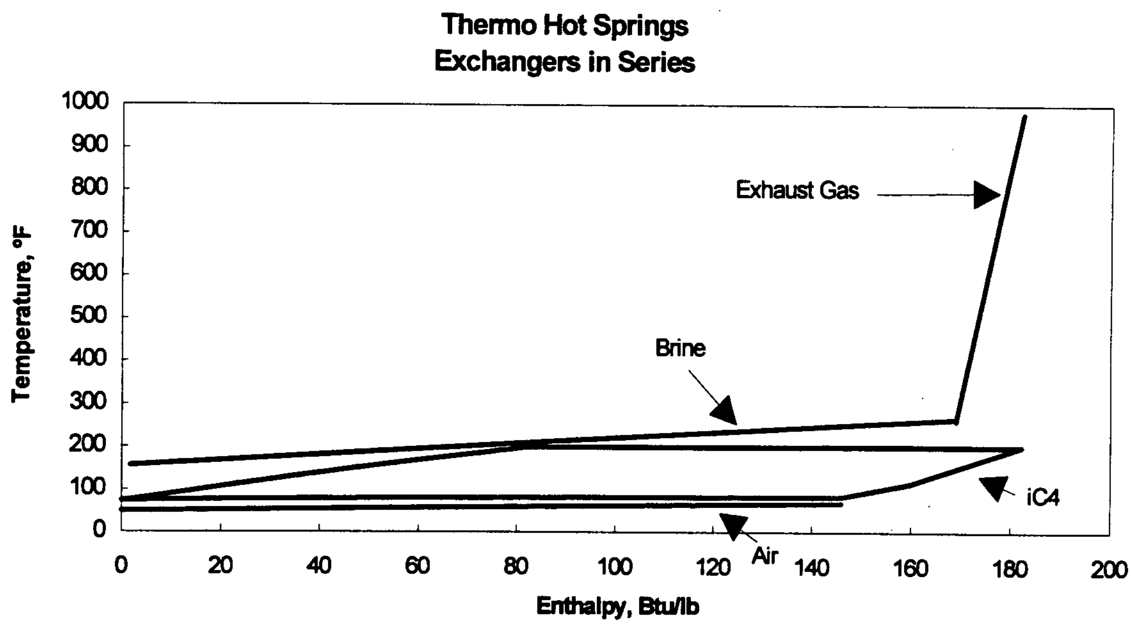

Figure 8-11
Temperature-Enthalpy Diagram: Heat Exchangers in Series

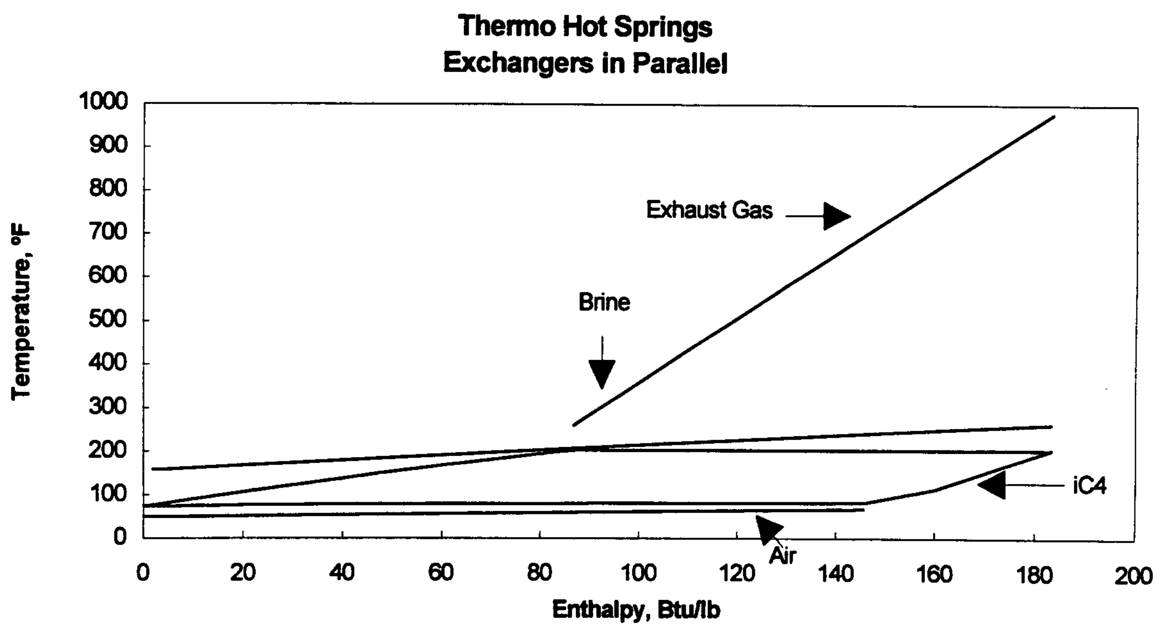

Figure 8-12
Temperature-Enthalpy Diagram: Heat Exchangers in Parallel

Performance Analysis. Gas turbine data are usually given at ISO conditions of 59°F and sea level. For this study, gas turbine power production, fuel consumption and exhaust gas flow were adjusted for the various site conditions.

Binary cycle assumptions were maintained the same as for the stand alone binary cycles. These assumptions are detailed in Section 5. Due to pressure drop requirements through the exhaust gas-to-brine heat exchanger, back pressure of 10 inches of water on the combustion gas turbine was used. Standard HRSGs used in combined cycle plants typically also have a pressure drop of 10 inches of water. The minimum temperature difference between the exhaust gas and the isobutane was set at 30°F and the minimum exhaust gas temperature was set at 260°F to limit condensation.

Economic Analysis. Binary cycle cost assumptions were same as those listed in Section 5. GE provided a budget price of $8,600,000 for the natural gas turbine package. A vendor quoted a price of $1,077,500 for an exhaust gas exchanger with a duty of 902 MMBtu/h and a weighted log mean temperature difference of 223 °F. An installation multiplier of 1.8 (refer to the section on dual flash/gas turbine hybrid cycles) was used for the combustion turbine and exhaust gas exchanger.

Results

Depending on site elevation and temperature, the combustion gas turbine produces between 18 and 19 MW. The binary cycle produces the remainder. Table 8-6 shows the specific output of the binary portion of the plant as well as that of a stand alone binary cycle plant. Addition of exhaust heat increases brine utilization 22% at Thermo Hot Springs, 25% at Raft River, 14% at Vale and 7% at Surprise Valley. Addition of high temperature exhaust gas has a larger impact on cycle efficiency for low temperature resources where thermodynamic efficiency is inherently lower. Table 8-7 shows percentage of heat input to the binary cycle that comes from the exhaust gas. At Thermo Hot Springs, the percentage of heat input from exhaust gas is lower than for Raft River so specific power output increases less at Thermo Hot Springs than at Raft River.

Table 8-6
Specific Output, kWh/1000 lb brine

	Resource Temp., °F	Hybrid Series Hx	Hybrid Parallel Hx	Binary
Thermo Hot Springs	265	2.69	2.99	2.44
Raft River, Idaho	300	5.04	5.05	4.04
Vale, Oregon	330	6.81		5.99
Surprise Valley, CA	375	9.10		8.49

Table 8-7
Exhaust Gas Heat Input

	% Heat Input from Exhaust Gas
Thermo Hot Springs	5.8%
Raft River, Idaho	8.6%
Vale, Oregon	9.8%
Surprise Valley, CA	9.6%

For low temperature resources, a plant with parallel heat exchangers has both higher specific power output and lower cost than one with a series heat exchanger arrangement. For low temperature cycles, the parallel arrangement allows a higher turbine inlet pressure. This is due to the shape of the vaporization curve for subcritical (boiling) cycles (See Figure 8-12).

For supercritical cycles (i.e. those above the critical temperature and pressure in which vaporization occurs without the formation of a distinct two-phase condition), heat is better utilized by adding the hottest heat source to the hottest portion of the cycle.

Table 8-8 shows the specific capital cost for both binary/natural gas hybrid plants and binary plants. Capital cost of hybrid plants is significantly lower due to the lower first cost of the combustion turbine. The gas turbine cost is about $865/kW. The economically preferred generation technology is determined by the levelized busbar cost which includes natural gas fuel costs. Levelized busbar costs for the various technologies are discussed in Section 10.

Table 8-8
Specific Capital Costs

	Resource Temp., °F	Hybrid Series Hx	Hybrid Parallel Hx	Binary
Thermo Hot Springs	265	$2,763	$2,726	$4,188
Raft River, Idaho	300	$1,961	$1,964	$2,945
Vale, Oregon	330	$1,704		$2,356
Surprise Valley, CA	375	$1,528		$2,115

9

ENVIRONMENTAL IMPACT

Introduction

Although geothermal power plants are usually environmentally benign compared to other energy resources, there are some differences between the various technologies that should be understood and evaluated. Since the overall environmental impact is usually low, this is a secondary criterion in comparing technologies.

Since all of the technologies are geothermal power plants, we can expect that they will all have roughly the same impact in a number of environmental areas. In this section, we first discuss those areas for which all technologies have roughly the same environmental impact. Next, we examine the areas for which different technologies have significantly different environmental impacts. Once we have evaluated the various environmental impacts, we can rank the technologies against each other.

Technology Independent Impacts

Earth

At ground level, construction of a geothermal power plant will change the topography of the site. Operations such as grading will serve to disrupt and displace the earth. Although geothermal plants are a clean alternative to many other forms of energy production, surface runoff containment is usually necessary in order to prevent unacceptable contamination of nearby waterways.

The effects of operating a plant may also be seen below ground. The brine is cooled and moved from the production well areas to the injection well areas. This movement of brine has the potential to cause subsidence, which could damage structures in the area. In some cases it is necessary to add chemicals to alter the pH of the brine to prevent scaling. This could then alter the pH of the reservoir.

All of the technologies we have investigated have roughly the same potential to impact the surrounding earth.

Plant and Animal Life

Construction of a geothermal power plant would, as would any kind of construction project, impact the plant and animal life in the area. It would result in the destruction of a small amount of habitat, and disrupt the lives of a few organisms. However, it is not inherent in any of the geothermal technologies to do the following: change the diversity of species, reduce the number of an endangered species, or introduce a new species to an area.

Noise

Each type of technology has pumps, turbines, cooling fans, vents, etc. All of these would contribute to a steady drone of background noise. One type of technology would not be significantly noisier than another. The degree of impact of this noise would vary with the proximity of the plant to human population. Often, geothermal sites are in remote regions, and concern with noise pollution by geothermal power plants is minimal.

Light and Glare

Geothermal power plants operate around the clock. It is therefore necessary that they be well lit to allow routine operation and inspection at night. This light would be noticeable in the surrounding area, and would not vary significantly from one technology to another. The impact of this light would be minimal since geothermal plants are usually located in remote regions.

Land Use

The main plant would take up from 5 to 10 acres of land. The gathering and injection systems would be spread over several miles in all directions. Several miles of transmission lines are usually required to deliver the electric power to consumers. Also, in some cases, roads must be built or expanded to accommodate construction traffic. For a 50 MW plant, this land use would not vary significantly from technology to technology at a given site.

Population

A power plant would increase population by adding to the area people who construct and then run the plant. This additional population would require housing, transportation, utilities, and public services. The degree to which this would impact an area would depend upon the existing population. All of the technologies would have roughly the same impact on population.

Recreation

A geothermal power plant might impact outdoor recreation activities. A branching gathering and injection system might serve as a barrier to hunters, hikers, or off road motorists. All of the technologies have roughly the same potential to impact recreation in these ways.

Archaeological/Historical

It is possible that the site chosen for a geothermal power plant could contain fossils, or ruins or other items of archeological interest. It is also possible that the sites contain structures or landmarks of historical interest. This type of potential impact would not be affected by the type of technology in use.

Technology Dependent Impacts

Air Pollution

Air pollutant emissions from geothermal power plants do vary from one technology to another. Open cycle plants emit noncondensable gas (NCG) produced with the brine into the air. The open cycle technologies include dual flash, advanced flash, sub-atmospheric flash, rotary separator turbine, and flash hybrids.

Refer to the standard dual flash process flow diagram Figure 5-9 to follow the path that the gas travels in a typical open cycle. The gas begins dissolved in the brine and then migrates from the brine into the steam when the brine is flashed. It then goes through the turbine and into the condenser. Some of the gas leaves the condenser dissolved in the liquid condensate and goes to the cooling tower with the cooling water, while the remainder is vented to the atmosphere, or, if required, sent to an H_2S abatement facility.

Hydrogen sulfide (H_2S) is a hazardous constituent of the noncondensable gas stream. It is colorless with a strong odor. It is heavier than air and will settle in low lying areas. Hydrogen sulfide exposures of 800 to 1000 ppm can be fatal in 30 minutes. In concentrations of 20 to 150 ppm, hydrogen sulfide acts as a respiratory irritant by combining with alkali present in moist tissues to form sodium sulfide, a caustic. In order not to exceed harmful ground level concentrations, it may be necessary to install an H_2S abatement facility. Typically, for a nominal 50 MW power plant this restriction is equivalent to reducing H_2S to about 100 grams/MW or 11 lbs/hr. In this study, we have included an H_2S plant whenever the H_2S release exceeded 11 lbs/m.

The amount of CO_2 emitted by an open cycle geothermal plant is directly proportional to the brine noncondensable gas content, and inversely proportional to brine utilization. In general, CO_2 emissions from geothermal plants measured in lb/kWh are significantly less than emissions from natural gas plants. Thus, geothermal plants when combined with natural gas plants will have the net effect of reducing CO_2 emissions compared to a natural gas only plant. In instances when the brine NCG content is high, this benefit will not be attained. However, even in such instances combining a geothermal plant with a natural gas plant will be environmentally beneficial. Natural gas plants emit harmful NO_x gases whereas geothermal plants do not. Therefore, NO_x emissions from a geothermal/natural gas hybrid will be lower than from a natural gas only plant.

Ammonia (NH_3), hydrogen (H_2), and methane (CH_4) are other common trace constituents in geothermal brine. In our economic analysis, we do not make any assumptions as to the concentrations of these species in the brine. Generally, the ammonia concentrations are less than that of H_2S but on the same order of magnitude. The others are found in one tenth the concentration of H_2S or smaller. The trends for emission of CO_2 reflect trends for emissions of these trace substances: improving brine utilization for a given technology at a given site reduces the relative amount of emissions.

All varieties of binary plants are closed cycle plants. Closed cycle plants do not expose the geothermal fluid to the atmosphere and therefore do not do not emit noncondensable gases. Flash binary hybrid II, the back-pressure turbine cycle, can also be a closed cycle plant.

Although these closed cycle plants do not emit noncondensable gases, they do emit some of their working fluids through leakage. Binary cycles emit mixtures of light hydrocarbons such as isobutane and isopentane, which are colorless gases that act as respiratory irritants. Data from operating binary plants shows that they lose some working fluid circulation. In general, colder resources use greater amounts of working fluid and therefore leak a greater quantity of working fluid.

The gas turbine portion of the gas turbine hybrid plants burns natural gas, which is predominantly methane. The normal combustion products are CO_2 and water. In addition to these products, side reactions involving the nitrogen and oxygen in the air produce nitrogen oxides, or NO_x. The gas turbine emissions table (Table 9-1) shows NO_x emissions for the two types of gas turbine hybrid plants.

Water Consumption

Water cooled plants that do not use condensate as cooling water makeup will require an outside water source. This can impact the surrounding domestic and natural water supply. In this study, only water cooled binary plants do not use condensate as make up.

Aesthetics

All of the technologies that are water cooled employ cooling towers. Cooling towers will have visible plumes of water vapor that vary in size depending upon the amount of water evaporated and on the humidity. Although these plumes

Table 9-1
Gas Turbine Exhaust

Site	Dual Flash - Gas Turbine		
	Fuel #/hr	NOx #/hr	NOx ppm
Coso Hot Springs	9,407	11.79	25
Desert Peak	9,476	11.87	25
Dixie Valley	9,879	12.38	25
Raft River	9,566	11.99	25
Glass Mountain	9,276	11.62	25
Salton Sea	10,700	13.41	25
Surprise Valley	9,476	11.87	25
Thermo Hot Sp	9,068	11.36	25
Vale, Oregon	9,625	12.06	25

Site	Binary - Gas Turbine		
	Fuel #/hr	NOx #/hr	NOx ppm
Raft River	9,564	11.98	25
Surprise Valley	9,479	11.88	25
Thermo Hot Sp	9,066	11.36	25
Vale Oregon	9,625	12.06	25

are essentially clean distilled water vapor, to some they may appear unattractive. Depending on the location of the plant, this can be of concern.

Safety

Conventional binary cycles use flammable working fluids at high pressures. They also receive regular shipments of working fluids to make up for leakage. These flammable components of the process pose a risk of explosion and fire not associated with other technologies that do not employ flammable working fluids.

Gas turbine hybrid plants burn natural gas at high pressures and at temperatures of over one thousand degrees Fahrenheit. They burn large quantities of gas that are on the order of ten thousand pounds per hour. These conditions also pose a fire risk not found in other geothermal technologies.

Rankings

Based on the evaluation of the environmental impacts that vary significantly from one technology to another as discussed above, we may rank the technologies against each other. We have ranked the technologies from first on down based on a point system, and tabulated the rankings in Table 9-2.

Table 9-2
Environmental Impact Rankings

		1					2	3	4	
		Air Emissions					Water	Aesth	Safety	
Rank	Technology	NCG CO_2	NCG H_2S	Fuel NOx	Fuel CO_2	Work Fluid	Fresh mk-up	Visible Plume	Fire Risk	Total Score
1	Binary (Mixed Fluids)	1	1	1	1	0	1	1	0	**6**
2	Binary (Air-cooled)	1	1	1	1	0	1	1	0	**6**
3	Dual Flash/Binary II	1	1	1	1	0	1	1	0	**6**
4	Rotary Sep Turbine	0	0	1	1	1	1	0	1	**5**
5	Sub-atm Flash	0	0	1	1	1	1	0	1	**5**
6	Advanced Flash	0	0	1	1	1	1	0	1	**5**
7	Dual Flash	0	0	1	1	1	1	0	1	**5**
8	Binary/Gas Turb	1	1	0	0	0	1	1	0	**4**
9	Binary (Water-cooled)	1	1	1	1	0	0	0	0	**4**
10	Dual Flash/Binary I	0	0	1	1	0	1	0	0	**3**
11	Dual Flash/Gas Turb	0	0	0	0	1	1	0	0	**2**

Each technology receives a score from zero to eight, eight being best. A technology can earn one point in each of eight categories, if it does not have that particular negative environmental impact. Air emissions encompass five of the eight ranking categories. Water use, plume visibility, and fire risk are the other three.

In the event of a tie score, we give a higher ranking to the technology with better brine utilization. In many of the areas where environmental impact is not dependent upon the technology chosen, a plant that has better brine utilization would tend have a lesser environmental impact by virtue of its reduced scale. A smaller gathering system, for example, would tend to lessen impacts on land use, animal life, and so on.

10

NEXT GENERATION GEOTHERMAL POWER PLANT TECHNOLOGIES

Concept Ranking Methodology

The goal of the next generation geothermal power plant (NGGPP) study is to identify the best alternative for geothermal power generation at each of the sites. The most attractive technology at each of the sites will generally be the technology with the lowest cost of power generation. Therefore, levelized busbar cost is the primary criteria for ranking the technologies. Since the costs developed in this study were based on installed capacity factors rather than detailed material takeoffs and since equipment quotes were not received for each item at each site for each technology, the uncertainty in absolute costs presented in this study is significantly greater than ± 5%. However, since a consistent set of assumptions and the same methodology was used for each case for comparative purposes, the costs are valid to at least ± 5%. For this reason, technologies with levelized busbar cost within 5% of each other are considered to be equivalent and specific output is used as a tie-breaker to determine the relative ranking of the technologies.

Specific Output

A technology with a higher specific output is more efficient and better utilizes the geothermal resource. A technology with higher specific output will use less brine per kWh of electricity produced, thereby extending the life of a finite resource. Table 10-1 lists the economic optimum specific output for each technology at each site studied. For technologies with specific output within 5% of each other, environmental impact is used to determine the relative ranking.

Environmental Impact

Technologies with minimal environmental impact are preferred. A technology with a lower environmental impact will be easier to permit; thereby, reducing project costs. In addition, by being perceived as a good neighbor, the utility will generate goodwill in the community. In the Western United States few geothermal resources are located near an abundant fresh water source, so water consumption is an important criteria in establishing a technology's environmental impact. Environmental impact of each of the technologies was discussed in detail in Section 9 and the environmental impact rankings are summarized in Table 9-2.

Levelized Busbar Cost Methodology

The levelized busbar cost of power generation was calculated using the methodology explained in EPRI's 1993 Technical Assessment Guide (TAG) for levelized revenue requirement (EPRI, 1993). The revenue requirement is defined as, "the total revenue that must be collected from customers to compensate a utility for all the expenditures associated with implementing a decision involving money." The levelized revenue requirement (LRR_A) for the power plant was calculated using TAG Equation 6-27.

LRR_A = (Investment)Pm,n + Σ (Expenses)Ln
where,

> LRR_A = Levelized Annual Revenue Requirement
> Investment = Total plant cost
> Expenses = All appropriate expenditures
> Ln = Levelizing factor, 1.541 (EPRI, 1993)
> Pm,n = Levelized carrying charge of a plant item with an m year tax recovery class and n year book life (EPRI, 1993)

Table 10-1
Specific Output, kWh/1000 lb Brine

Resource	Salton Sea	Coso	Glass Mtn.	Dixie Valley	Desert Peak	Surprise Valley	Vale	Raft River	Thermo Hot Spr.
Resource Temperature, °F	570	525	510	450	425	375	330	300	265
Baseline Technologies									
Air-cooled Binary			13.06			8.49	5.99	4.04	2.44
Water-cooled Binary						7.45	5.80	3.90	2.15
Com'l Dual Flash (DF)	18.40	16.25	17.11	11.98	10.65	6.69	4.64	3.06	
Advanced Binary Cycles									
Mixed Fluids (MF)						7.09	5.26	3.52	2.18
Binary-Synchronous Turbine			13.47			8.79	5.97	4.17	2.58
MF-Synchronous Turbine						8.10	5.57	3.78	2.25
Metastable Expansion						7.17	5.67	4.00	
Advanced Flash									
RST	19.58	16.43	17.28	12.05	10.71				
Adv Flash, Reheater	18.58		17.12	11.88	10.57	6.51			
Sub-Atm Flash						6.74	4.70	3.13	
Hot Water Turbine			17.94	12.74				3.16	
Geothermal Hybrid Cycles									
DF/ Bin. Bottoming			19.44		12.15	7.86	5.17	3.88	
DF/Back Pressure Turbine		15.55	17.36		10.41	7.42	4.95		
Hot Dry Rock						19.94			

The assumptions listed in Table 10-2 have been obtained from TAG Table 6-3. These assumptions were used by EPRI to calculate that Pm,n is 0.141 and Ln = 1.541 (EPRI, 1993).

Table 10-2
Levelized Busbar Cost of Power Generation Assumptions

Tax Recovery Period	5 years
Book Life	30 years
Discount Rate	9.2%/year (after tax)
Inflation Rate	4.1%/year
Federal and State Income Tax	38%
Property Tax and Insurance	2%

The investment consists primarily of the capital cost for the plant and the gathering system. Table 10-3 lists the specific cost in $/kW for each of the technologies at each of the sites studied. The assumptions employed in calculating the capital cost of a 50 MW geothermal plant were detailed in Section 4.

Other costs included in the investment cost are listed in Table 10-4. Owner's costs include an allowance of $65/kW for siting and licensing, $100/kW for financing, and $100/kW for project costs. Resource development costs listed in Table 10-4 are also included as part of the initial capital investment. A capacity factor of 96% was used for all technologies. Existing binary and dual flash plants operate with capacity factors in excess of 96%. Emerging technologies will need to achieve capacity factors of 96% if they are to be commercial.

Expenses consist of operating and maintenance costs, chemical costs, fuel costs, well rework, and royalty payments. Table 10-5 summarizes the assumptions used to calculate these.

Annual operating and maintenance costs were assumed to be $95/kW for all geothermal concepts. This value is based on review of operating costs at existing binary and dual flash plants. O & M costs include operating labor for both the plant and the wellfield, maintenance, equipment and materials, spares, office supplies and equipment, plant vehicles, building and ground maintenance, utilities and plant management. It also includes some allowance for overhead and contingency. An O & M cost of $71.25/kW was used for the natural gas hybrid cycles, because the relatively smaller well field would require less maintenance.

Table 10-3
Specific Cost, $/kW (1)

Resource	Salton Sea	Coso	Glass Mtn.	Dixie Valley	Desert Peak	Surprise Valley	Vale	Raft River	Thermo Hot Spr.
Resource Temperature, °F	570	525	510	450	425	375	330	300	265
Baseline Technologies									
Air-cooled Binary			2,070			2,115	2,356	2,945	4,188
Water-cooled Binary						2,015	2,302	2,869	4,161
Com'l Dual Flash (DF)	1,057	1,588	1,504	1,225	1,491	2,242	2,634	3,161	
Advanced Binary Cycles									
Mixed Fluids (MF)						1,896	2,207	2,701	3,919
Binary- Synchronous Turb.						1,817	2,346	2,841	3,924
MF - Synchronous Turbine						1,770	2,072	2,561	3,633
Metastable Expansion						2,157	2,464	2,925	
Advanced Flash									
RST	1,083	1,608	1,523	1,235	1,513				
Adv Flash, Reheater	1,070		1,514	1,246	1,511	2,325			
Sub-Atmospheric Flash						2,238	2,631	3,108	
Hot Water Turbine			1,537	1,280				3,318	
Geothermal Hybrid Cycles									
DF/ Bin. Bottoming			1,635		1,544	2,129	2,669	3,068	
DF/ Back Pressure Turbine		2,137	2,062		2,151	2,565	2,983		
GT/ Binary Hybrid						1,528	1,704	1,961	2,726
GT/ Dual Flash Hybrid	1,116	1,510	1,405	1,195	1,342	1,641	1,902	2,194	3,946
Hot Dry Rock						8,900			

(1) Valid only for future plants estimated using the cost assumptions given in this study.

Although chemical usage does not have a significant impact on the levelized busbar cost, it was identified as a separate line item since different plant cycles will have different chemical usage. For binary cycles, a chemical cost of $60,000/year was included for working fluid makeup and miscellaneous chemical usage. For water-cooled plants, $80,000/year was added for cooling water treatment. Actual water treatment costs will vary with site conditions and with the composition of the water used for cooling tower makeup. A cost of $147/long ton of sulfur was included for the chemicals used in the liquid redox hydrogen sulfide abatement system. For all power generation concepts, $0.284 annually per lb/h brine or 0.17¢/kWh for a 50 MW plant with a brine flow of 2,600,000 lb/h was added for pH modification of the hypersaline Salton Sea brine. This cost was based on the information that 100-120 ppm of HCl is added at Salton Sea. (Hoyer, Gallup and Kitz, 1991)

Table 10-4
Levelized Busbar Cost of Power Plant Assumptions

Capacity Factor	96%
Investment Tax Credit	10%
Owner's Costs	$265/kW
Resource Development Costs	Thousand Dollars
Soft Cost/Investment	
Geologic Mapping	90
Geochemistry	95
Gravimetry	90
Geoelectrical Survey	125
Temp. Gradient Drilling	335
Modeling	65
Exploratory Investment	
Primary Test Well	2500
Secondary Test Well	1875

Table 10-5
Operating and Maintenance Cost Assumptions

O & M Costs	$95/kW/year
Chemical Cost	$60,000/year for binary cycles
	$80,000/year for dual flash cycles
	$33/MMlb brine for Salton Sea
Fuel Costs	$4.36/MMBtu (EPRI's prediction)
	$1.73/MMBtu (current price 12/1/94)
Well Rework	2% of initial well cost annually
Royalty	10% of revenue requirement

Natural gas prices are quite variable and long term market trends are difficult to predict. EPRI's 1993 TAG predicted a natural gas price of $4.36/MMBtu based on the higher heating value. The current spot price is actually $1.73/MMBtu. Because the levelized revenue requirement for a combustion turbine are very dependent on natural gas price, levelized revenue requirement was calculated for both gas prices. The values presented in the final table of levelized revenue requirement are the average of these two gas prices.

Table 10-5 gives a value of 2% of the initial well cost annually for well replacement and rework. This value, which was provided by EPRI, was used in our study, but it is probably low. The actual cost of well maintenance is site specific. This study assumes that a royalty of 10% of revenue is paid to the lease holder.

Levelized Busbar Cost Results

Table 10-6 lists the levelized revenue requirement for power generation for each of the technologies at each of the sites. It is important to note that these revenue requirements are intended to allow comparison of competing technologies on an equal basis. The absolute values are valid within the relatively narrow context

Table 10-6
Levelized Revenue Requirement, ¢/kWh [1]

Resource	Salton Sea	Coso	Glass Mtn.	Dixie Valley	Desert Peak	Surprise Valley	Vale	Raft River	Thermo Hot Spr.
Resource Temperature, °F	570	525	510	450	425	375	330	300	265
Baseline Technologies									
Air-cooled Binary			6.34			6.33	6.77	7.90	10.22
Water-cooled Binary						6.22	6.71	7.80	10.30
Com'l Dual Flash	4.69	5.42	5.29	4.83	5.38	6.63	7.37	8.51	
Advanced Binary Cycles									
Mixed Fluids						5.92	6.53	7.53	9.83
Binary- Synchronous Turb.			5.77			5.83	6.76	7.72	9.78
M.F. - Synchronous Turbine						5.77	6.34	7.30	9.35
Metastable Expansion						6.42	6.99	7.87	
Advanced Flash									
RST	4.71	5.43	5.30	4.83	5.27				
Reheater	4.81		5.30	4.88	5.43	6.85			
Sub-Atmospheric Flash						6.60	7.43	8.36	
Hot Water Turbine			3.54	4.92				8.68	
Geothermal Hybrid Cycles									
DF/ Bin. Bottoming			5.43		5.18	6.44	7.38	8.26	
DF/ BP Turbine		6.43	6.17		6.53	6.94	7.48		
GT Hybrid Cycles									
GT/ Binary Hybrid						6.35	6.34	6.78	8.01
GT/Dual Flash Hybrid	6.36	6.65	6.29	5.46	6.61	6.52	6.90	7.21	10.17
Hot Dry Rock						19.94			

(1) Valid only for future projects using the costs and economic assumptions given in this study.

of the economic assumptions used in this study. Specifically, they are appropriate for new projects only and should not be used to evaluate existing projects at any of the sites since the costs and other economic parameters are certain to be different from those assumed in this study.

Trends visible in this table have been discussed throughout this report. The cost of power generation usually increases with decreasing resource temperature. For the hotter self flowing wells, some reservoir specific characteristics can cause costs to be above or below this trend.

For example, plants at Coso (525°F) have a slightly higher cost than Glass Mountain (510°F) due to the higher noncondensable gas content of the steam at Coso. Also, power generation at Dixie Valley (450°F) is less expensive than at Glass Mountain due to the higher well productivity at Dixie Valley. Brine flow per well and other plant characteristics are similar for all four of the cooler sites, so specific cost monotonically decreases with resource temperature.

For the baseline technologies, dual flash plants are preferred for higher temperature resources and binary plants for lower temperature resources. The transition from dual flash to binary would be at some temperature between 375°F to 425°F. Cost of onsite plant equipment for a flash plant is significantly less expensive than for a binary type plant. With a higher temperature resource, well costs are a lower percentage of the total cost, so a simpler, less efficient plant gives the best economic performance. With colder resources, well costs increase dramatically, so a more expensive but more efficient plant is needed.

Table 10-7 shows the levelized revenue requirement for the natural gas cycles at three different natural gas prices.

Concept Ranking Results

Table 10-8 summarizes the relative ranking for each of the technologies at each of the sites. A discussion of the results follows.

Salton Sea

The levelized revenue requirement for power generation using a baseline dual flash plant at Salton Sea is 4.69 ¢/kWh.

In this study, turbine inlet temperature was limited to 360°F due to metallurgical concerns. The optimum high pressure flash temperature for a dual flash plant operating at Salton Sea is higher than 360°F. As a result, RST is able to generate 7% more power than a conventional dual flash plant at nearly the same busbar cost of power generation.

Table 10-7
Levelized Revenue Requirement for Natural Gas Cycles, ¢/kWh [1]

Resource	Salton Sea	Coso	Glass Mtn.	Dixie Valley	Desert Peak	Surprise Valley	Vale	Raft River	Thermo Hot Spr.
Resource Temperature, °F	570	525	510	450	425	375	330	300	265
Natural gas at $4.36/MMBtu									
Gas Turbine/Dual Flash	7.41	7.57	7.15	6.24	7.59	7.35	7.66	7.96	10.89
Gas Turbine/Binary						7.21	7.13	7.56	8.75
Natural gas at $1.73/MMBtu									
Gas Turbine/Dual Flash	5.32	5.73	5.42	4.68	5.62	5.68	6.14	6.45	9.46
Gas Turbine/Binary						5.48	5.55	6.00	7.27
Average									
Gas Turbine/Dual Flash	6.36	6.65	6.29	5.46	6.61	6.52	6.90	7.21	10.17
Gas Turbine/Binary						6.35	6.34	6.78	8.01

[1] Valid only for future projects using the costs and economic assumptions given in this study.

Table 10-8
Concept Ranking

Resource	Salton Sea	Coso	Glass Mtn.	Dixie Valley	Desert Peak	Surprise Valley	Vale	Raft River	Thermo Hot Spr.
Resource Temp., °F	570	525	510	450	425	375	330	300	265
Baseline Technologies									
Air-cooled Binary	—	—	5	—	—	4	2	3	4
Water-cooled Binary	—	—	—	—	—	7	6	4	6
Dual Flash	2	1	2	2	2	8	10	8	—
Advanced Binary Cycles									
Mixed Fluids	—	—	—	—	—	3	3	5	5
Binary-Synch. Turb.	—	—	3	—	—	1	2	3	2
MF -Synch. Turbine	—	—	—	—	—	2	1	2	3
Metastable Expansn	—	—	—	—	—	6	5	3	—
Advanced Flash									
RST	1	1	2	2	2	—	—	—	—
Adv Flash, Reheater	2	—	2	2	2	10	—	—	—
Sub-Atm Flash	—	—	—	—	—	8	10	8	—
Hot Water Turbine	—	—	2	1	—	—	—	8	—
Hybrid Cycles									
DF/ Bin. Bottoming	—	—	1	—	1	5	9	7	—
DF/Back-Press.Turb.	—	2	4	—	3	8	8	—	—
GT/ Binary Hybrid	—	—	—	—	—	9	4	1	1
GT/ Dual Flash Hyb.	3	3	6	2	4	11	7	6	7

Salton Sea is a hypersaline, highly corrosive resource. Brine handling is a special concern for any plant installed on this resource. In this study, yearly levelized chemical cost for pH control of about 0.3 cents/kWh was added for all technologies. Excessive scaling or material compatibility may be a concern with RST operating on this resource.

The flash reheater concept is a variation on dual flash. Based on the ranking criteria of this study, flash reheater and dual flash have the same ranking since their cost and specific output are within 5%. However, since the cost of the flash reheater cycle is higher and the specific output is lower than dual flash cycles, it is unlikely that the incentive to install the more complex, higher risk flash reheater plant in preference to a dual flash plant will exist.

Coso

Due to the higher noncondensable gas content at Coso, some of the dual flash alternatives were not appropriate. Conventional dual flash and RST have nearly identical levelized cost and efficiency. One of the assumptions of this study was that the wellhead pressures and flows were the same for all the wells at a given site. Consequently, the RST technology did not offer a significant advantage for the power plant as a whole. Currently, however, a DOE sponsored test of RST is occurring at Coso on one well that has a significantly higher pressure than other wells feeding the high pressure steam manifold. For resources in which there is significant variation in wellhead pressures, the RST technology may offer a way to increase the overall performance of a steam flash power plant by recovering energy that would otherwise be lost in throttling the steam to the lowest common pressure.

At a resource with higher noncondensable gas content and higher H_2S content in the steam, the next generation geothermal technology is not likely to be a novel cycle. Rather, development should focus on more efficient gas removal systems and more cost effective sulfur handling systems.

Glass Mountain and Desert Peak

At Glass Mountain, a dual flash/binary bottoming cycle is 2.6% more expensive, but 14% more efficient than dual flash. At Desert Peak, a dual flash/binary bottoming cycle is both less expensive and more efficient than dual flash. Both Desert Peak and Glass Mountain have relatively low flow from each well. This makes well and gathering system costs relatively more expensive. As a result, it is important to achieve a higher thermal efficiency than is possible in a straight dual flash cycle.

This study assumes that all plants produce a nominal 50 MW net. Since hybrid plants are essentially two smaller plants with some heat transfer between the two plants, economies of scale tend to make the hybrid plants more expensive than non-hybrid plants. Even so, dual flash/binary is the most cost effective option at Desert Peak.

Dual flash/binary bottoming cycle is a relatively low risk advanced technology. The dual flash component is identical to a standard dual flash plant. Similarly, the binary portion is a standard air-cooled binary plant. The heat exchanger that couples the plants is identical in design to the brine/hydrocarbon exchanger in a binary plant. In the design of this plant, attention must be paid to the scaling properties of brine at lower temperatures.

Dixie Valley

Since the hot water turbine has 2% higher cost but more than 6% better brine utilization, the hot water turbine is the preferred technology at Dixie Valley. Commercial dual flash, rotary separator turbine and dual flash with a reheater all have nearly identical levelized revenue requirement and specific output. Since there is no incentive for the higher first cost options of RST or dual flash with an added reheater, commercial dual flash is preferred to these two options at Dixie Valley.

Surprise Valley and Vale

At these warmer pumped well sites, the levelized revenue requirement for many of the advanced binary concepts are similar. The synchronous turbine proposed by Barber-Nichols is higher efficiency and lower cost than the turbines currently used for binary cycles. Therefore, the synchronous turbines do consistently better in both cost and efficiency than competing cycles without them.

At Vale, the mixed fluid cycle has 7% lower cost but 20% lower specific output than a binary cycle using isobutane. A mixed fluid cycle plant at Surprise Valley would consist of major equipment identical to the equipment used in commercial binary plants but the working fluid would be 96% isobutane and 6% heptane instead of 100% commercial isobutane. A mixed fluid cycle with a synchronous turbine is ranked as the preferred technology at Surprise Valley.

At Surprise Valley, mixed fluid cycle is less than 5% lower cost than an isobutane cycle on this reservoir. Since the isobutane cycle has 14% better brine utilization, a synchronous turbine on a standard binary cycle is the preferred technology.

Water-cooled binary cycles are 1 to 2% more cost effective than air-cooled binary cycles, but these water-cooled cycles also have lower specific output.

Raft River and Thermo Hot Springs

Power generation from these cooler resources requires an efficient conversion technology to be cost effective. A gas turbine/binary hybrid is an example of such a technology. Technical risk associated with this hybrid is relatively low. The performance and reliability of standard commercial gas turbines are well documented and binary geothermal plants are also a proven technology. The exhaust gas heat exchanger which couples the fossil fuel and geothermal cycles is similar to waste heat recovery units in cogeneration facilities.

The primary risk associated with the gas turbine hybrid option is natural gas price. Using EPRI's price for natural gas of $4.36/MMBtu, the hybrid is not the low cost alternative. Using today's price of $1.73/MMBtu, a natural gas hybrid is the most cost effective generation method at both Raft River and Thermo Hot Springs. In recent years, predictions of fuel price increases have usually been high. For assigning the ranking, busbar cost based on the average of the higher and lower fuel prices was used.

A gas turbine hybrid would have significantly higher emission than other geothermal concepts. Although natural gas is the cleanest fossil fuel, emissions from a gas turbine hybrid are still significantly higher than with an all geothermal plant. (See Section 9).

At Raft River, power generation with a gas turbine/binary hybrid costs only 7% less than with a mixed fluid binary cycle with a synchronous turbine. At Thermo Hot Springs, the cost of power generation with a gas turbine/binary hybrid assuming an average fuel price is 10% lower than the cost of power generated with the most cost effective binary cycle. Therefore, a gas turbine/binary hybrid is the number one ranked technology at the two coldest geothermal resources.

11

CONCLUSIONS AND RECOMMENDATIONS

Section 10 set forth the preferred technologies for each of the resources. This section presents our conclusions and recommendations with respect to further development of hydrothermal resources. None of the technologies, even if all were successfully demonstrated, constitute a real breakthrough in terms of a dramatic reduction in the cost of generating power. The basic reason is that present designs are already thermodynamically efficient and the proposed designs are facing limitations imposed by the second law of thermodynamics. The best opportunities for dramatic cost reduction may lie in finding high temperature resources and in reducing the cost of drilling, subjects not of concern in this work.

We can expect incremental improvements in drilling, development and power generation leading to reduction in cost of 10 to 15%.

Before presenting our conclusions for specific technologies a general discussion of dispatchability and economies of scale as these relate to the NGGPP is presented below.

Dispatchability of power plants is a desirable characteristic for the utility companies. From a technical standpoint alone geothermal power plants can be operated at reduced loads. Steam flow to the axial flow turbines in a flash plant can be throttled in order to reduce power plant output. Similarly the working fluid flowrate can also be reduced in a binary cycle plant. Plant operators usually avoid shutting in self-flowing wells if reduced load operation is temporary. Steam is vented to the atmosphere or bypassed around the turbine and condensed in the condenser. If the plant is designed for steam to bypass the turbine, the condenser must be rated for higher temperature and duty.

While technically feasible, operation of a geothermal plant as a peaking plant rather than as a base-load plant is not economically feasible. For all geothermal plants regardless of technology, nearly all the levelized cost of power generation is return on capital investment. By contrast, in fossil fuel fired power plants fuel cost is a sizable fraction of the cost of power generation. At reduced load, fuel consumption is reduced making the load following capability of fossil fuel plants more economical than that of geothermal power plants. None of the technologies that were studied would materially improve the dispatchability of geothermal power. Gas turbine hybrid cycles would be more economical to dispatch than all-geothermal cycles for the foregoing reasons.

This study has used a 50 MW (net) power plant as the basis to evaluate various technologies for the NGGPP. Larger capacity plants would be more economical due to economies of scale. Development and engineering costs are almost independent of plant capacity. Moreover, the cost of major equipment scales up as a fractional power of the size ratio. Thus, the specific capital cost for larger capacity plants would be lower.

However, in developing the NGGPP it must be remembered that the capacity of geothermal power plants is constrained by the thermal capacity of the resource. The power generation capacity of the plant should be such that the resource can support power production over the entire life of the project. Since long-term performance of a reservoir is often not fully characterized in the early stages of development, smaller capacity plants lower the project's risk.

Our conclusions and recommendations for specific technologies are as follows.

Air-cooled Commercial Binary

Air-cooled commercial binary provides higher specific outputs and only slightly lower levelized cost of power generation than water-cooled commercial binary for the four resources evaluated. These four resources had essentially the same relatively cool, atmospheric condition. This result may not be duplicated at sites with higher atmospheric temperature. Also an assumption of this study is a non- time-differentiated cost of power. A water-cooled binary plant may be more cost effective for a summer peaking utility than an air-cooled binary plant providing that adequate make-up water is available providing that adequate make-up water is available.

Mixed Fluids

The use of mixed fluids shows a small but consistent reduction in power cost over the air-cooled commercial binary (up to 7%). Brine utilization suffers as much as 20% for the economic optimum mixed fluids cycle. We think it likely that the use of mixed fluids will become commonplace and that future binary plants will incorporate mixed working fluids. It is a lower risk modification requiring little or no development expense.

Synchronous Turbine

This design, proposed by Barber-Nichols, indicates a lower cost (from 4% to 8.5%) in three out of four cases when compared to the commercial air-cooled binary case. Since these synchronous turbines are reported to be more efficient and less expensive than the turbines used for the baseline binary cycles, cycles with the Barber-Nichols turbines consistently outperform those cycles with the baseline turbine. Building and operating a full scale demonstration module will be required to verify cost and performance.

Metastable Expansion

For a binary cycle plant operating with a geothermal fluid temperature of 330°F or greater, metastable expansion may be used to mitigate the impact of a declining resource. As the geothermal fluid temperature drops below design, the isobutane cycle can be operated closer to the two phase region allowing more energy to be extracted from the geothermal resource. Metastable expansion may be cost effective given this scenario if geothermal fluid does not scale excessively at the lower outlet temperature and if turbine efficiency is maintained at the new operating conditions.

Erosion of the turbine wheel during long term operation is a concern with this technology. INEL is currently investigating metastable expansion. With a fixed geothermal outlet temperature, this technology shows no cost advantage over the commercial air-cooled binary.

Rotary Separator Turbine

This technology shows no improvement or marginally lower cost (0.01 ¢/kWh to 0.02 ¢/kWh) and marginally improved performance in four out of the five applications. At Salton Sea where the thermodynamic optimum flash pressure is higher than allowed due to metallurgical limits on the turbine inlet temperature, RST has 6.4% improved brine utilization. As discussed in the body of this report, it has application where the wellhead pressure significantly exceeds the maximum allowable turbine inlet pressure. This condition can occur in a resource where one or more wells operate at a pressure significantly higher than the lowest wellhead pressure. When cost and performance have been verified on a commercial scale, the RST could have widespread application.

Reheater

Since the cost of the reheater itself is a small fraction of the total plant cost, this concept shows only marginally higher cost than commercial dual flash. However, addition of a reheater does not improve the specific output of a dual flash cycle, so there is little justification for this additional capital expenditure. Accordingly, there seems to be no reason to develop this concept further.

Sub-atmospheric Flash

Sub-atmospheric flash was considered as a concept that may make dual flash plants competitive with binary plants at lower resource temperatures. The power cost associated with this technology is significantly higher than the corresponding commercial air-cooled binary (0.27 ¢/kWh to 0.66 ¢/kWh) for the three cases studied (Surprise Valley, Raft River and Vale). Accordingly, there seems to be no reason to develop this concept further.

Hot Water Turbine

At Glass Mountain and Dixie Valley, installing a hot water turbine will increase the cost of power generation by 1 to 2%, but it will increase brine utilization by 5 to 6%. This technology is promising especially if the cost of the hot water turbine could be reduced.

Dual Flash/Binary Bottoming

This cycle shows no cost saving over either commercial binary or commercial dual flash for Glass Mountain, Surprise Valley, Vale or Raft River. A small saving (0.2 ¢/kWh) is indicated for Desert Peak. At both Glass Mountain and Desert Peak, dual flash/binary bottoming has 14% better specific output than dual flash. Thus we may conclude that this technology may be justified in particular cases.

Brine outlet temperature was taken from Table 4-1 based on the scaling potential of each brine. At Glass Mountain, Surprise Valley, Vale and Raft River the minimum brine outlet temperature was 150°F. At Desert Peak, the minimum temperature was 185°F. In many cases, silica scaling in the brine/working fluid exchangers will probably occur unless scale prevention measures are taken such as pH control, addition of inhibitors and the like. Laboratory and pilot scale testing to control scale is required on each potential application.

DF Back-Pressure-Turbine

This cycle shows higher costs ranging from 0.88 ¢/kWh to 1.15 ¢/kWh for the five cases studied (Coso, Glass Mountain, Desert Peak, Surprise Valley and Vale). This cycle is not competitive with an optimized cycle (either binary or flash) over a wide range of resource conditions. This higher cost represents the penalty from installing a closed, environmentally attractive system at a high temperature resource that would normally utilize the open dual flash cycle.

Hot Dry Rock

This technology shows a very high cost (almost 20 ¢/kWh), principally because of the high cost of wells. It appears that HDR is not yet competitive with power generated from hydrothermal resources.

Gas Turbine Hybrid Cycles

Using an average natural gas price, Gas Turbine Hybrid cycles are the lowest cost alternative at the two coldest resources, Raft River and Thermo Hot Springs. At the current natural gas price, a gas turbine/binary cycle plant is also cost effective at Vale. Cost effectiveness of the gas turbine hybrid cycles is very dependent on natural gas prices. One would expect that the levelized cost of power generation for gas turbine hybrids would be less than for the baseline technologies at all sites, not just Raft River and Thermo Hot Springs since 100% gas turbine power plants at present gas price can usually produce power at lower cost than geothermal plants. Gas turbine hybrid cases are high since reverse economies of scale lead to high plant costs which override the expected cost savings.

12

REFERENCES

Baumann, K., "Some Recent Developments in Large Steam Turbine Practice", *J. Inst. Elec. Eng.*, **59**, p. 581, 1921.

Bliem, C.J. and Mines, G.L., "Condensation Behavior of Supersaturated Turbine Expansions", *DOE Geothermal Energy R&D Program, Energy Conversion Program Review XI*, p. 177, 1992.

Bliem, C.J., *Private Communication to John Mortz*, 1994a.

Bliem, C.J, *Private Communication to John Mortz*, 1994b.

Bliem, C.J. and Mines, G.L., "Supercritical Cycle Investigations", *DOE Geothermal Energy R&D Program, Energy Conversion Program Review XI*, p. 181, 1992.

Bliem, C.J. and Mines, G.L., "Advanced Binary Geothermal Cycles", *The Final Report on the Heat Cycle Research Program of the U.S. Department of Energy* (draft copy), 1993.

Cerini, D. J. and Hays, L. G., "Power Production From Geothermal Brine With the Rotary Separator", *Proceedings of the 15th IECEC*, American Institute of Aeronautics and Astronautics, **2**, p. 788, 1980.

Cerini, "Demonstration of a Rotary Separator for Two-Phase Brine and Steam Flows", U. S. DOE, TID-28519, January 1978.

Demuth, O.J. and Kochan, "Analysis of Mixed Hydrocarbon Binary Thermodynamic Cycles for Moderate Temperature Geothermal Resources Using Regenerative Techniques", *Proceedings of the 17th IECEC, IEEE*, **2**, p. 1104, 1982.

Demuth, O.J., "Analysis of Mixed Hydrocarbon Binary Cycles for Moderate Temperature Geothermal Resources", *Proceedings of the 16th IECEC*, American Society of Mechanical Engineers, **2**, p. 1316, 1981.

Department of the Navy, *Engineering Weather Data*, Department of the Navy Naval Facilities Engineering Command, 1978.

DiPippo, R., "Geothermal Energy as a Source of Electricity", published by U.S. Department of Energy, p. 1, 1980.

DiPippo, R. and Vrane, D.R., "A Double-Flash Plant with Interstage Reheat", *GRC Trans.*, **15**, p. 381, 1991.

Duchane, D.V. (Program Manager), *Hot Dry Rock Geothermal Energy Development Program 1993 Annual Report*, US Department of Energy, 1994.

Duchane, D., *Private Communication to John Mortz*, 1994.

EPRI, Chapter 6 in "TAG™ Technical Assessment Guide", Volume I, Electricity Supply, 1992

EPRI, *Proceedings of the Third Annual Geothermal Conference and Workshop*, EPRI WS-79-166, 1979.

Fridleifsson, I.B. and Freeston, D.H., "Geothermal Energy Research and Development", *Geothermics*, **23** (2), pp. 175-214, 1994.

Gallup, D.L., " 'BIOX' - A New Hydrogen Sulfide Abatement Technology for the Geothermal Industries", *Geotherm Science & Technology*, **4** (2), pp. 129-144, 1994.

Gas Turbine World, *The 1990 Handbook*, **12**, 1990.

Gogh, Frank van, *Private Communication to Michael Walker*, 1994.

Hays, L., *Private Communication to John Mortz*, 1994.

Holt, Ben, *Proceedings of the Third Annual Goethermal Conference and Workshop*, EPRI WS-79-166, 1979.

Holt, B., Sims, A.V. and Campbell, R.G., *Proceedings of the Fifth Annual Geothermal Conference and Workshop*, EPRI AP-2098, 1981.

Hoyer, D., Kitz, K. and Gallup, D., " Salton Sea Unit 2 Innovations and Successes", *GRC Trans.*, **15**, p. 355, 1991.

Johnson, J.E., Tzap, S.J., Kelley, R.E. and Laczko, L.P., "Options Narrowed to Claus or Redox Processes", *Oil and Gas Journal*, Nov. 22, p.70, 1993.

Kohl, A.L. and Riesenfeld, F.C., "Gas Purification", 4th Edition, Gulf Publishing Company, p.521, 1985.

Mines, G.L. and Bliem,C.J., "Improving the Efficiency of Binary Cycles", *Geothermal Program Review VI Proceedings*, U.S. DOE, p.81, 1988.

Mines, G.L., "Field Investigations Examining the Impact of Supersaturated Vapor Expansion on Turbine Performance", *Geothermal Program Review XII Proceedings*, U.S. DOE, 1994.

Mortz, J.R., *Private Communication to Carl Bliem*, 1994.

Mortz, J.R., *Private Communication to Carl Bliem*, 1994.

Nishioka, R. and Kato, Y., "Reaction Type Hot Water Turbine", *GRC Trans.*, **18**, p.549, 1994.

Purohit, G.P., "Estimating Costs of Shell and Tube Heat Exchangers", *Chemical Engineering*, August 23, p.56, 1983.

Tiemat, B.W., Rie, H., Laird, A.D.K. and Zhao, S., "Condensation Film Coefficients for Mixtures of Isobutane and Isopentane", *Proceedings of the 15th IECEC*, American Institute of Aeronautics and Astronautics, **2**, p. 455, 1980.

Weres, O, "The Partitioning of Hydrogen Sulfide in the Condenser of Geysers Unit 15", *Geothermics*, **12** (1), pp. 1-15, 1981.

APPENDIX A

CALCULATION METHODOLOGY

The following materials in Appendix A are a general discussion of process calculations used for analyzing various NGGPP concepts.

Gathering System Spreadsheet

The size and cost of a well head gathering system depends on the site location and topography. For example it depends on whether the wells are located uphill or downhill from the plant, and on how far away the wells are from the plant. These are detailed engineering considerations and for the purposes of this study, a generic gathering system design was used to estimate the gathering system cost. This design lays out wells in a hexagonal pattern, and then collects them in a straight line. Up to 4 or 5 tiers were allowed in this configuration, and a fixed separation of 600 feet was used between wells and a distance of 750 feet was used between nodes.

Individual lines are sized based on velocity and/or pressure drop. Line sizes and lengths are used to estimate the cost of the gathering system. The magnitude of this cost does not affect the levelized cost of power production significantly since it represents only 1-2% of the total facility cost.

Well Pump Spreadsheet

Holt has developed a well pumping model based primarily on well depth, pump setting distance, well drawdown, and required pressure at the surface. The number of stages, the stage efficiency, and the motor efficiency is used to determine the cost of the pump and the required brake horsepower.

Parasitics

Air cooler fan horsepower is a significant parasitic load for the plant. It depends on how much air is required to perform the condensation of the working fluid. Holt's database which contains design parameters for vendor-quoted air condenser designs was used to estimate air-cooler cost and horsepower requirements.

Well pumping parasitics are based on the Holt well pump model described above. On a case by case basis, the motor horsepower required is input into the cost model. The cumulative parasitics are calculated by summing the number of wells required in order to estimate this parasitic load.

Working fluid pumping parasitics are calculated based on standard pump equations using the circulation rate of hydrocarbon, its density, the pressure losses in the Brine/Hydrocarbon heat exchange train, and the pressure required at the turbine inlet.

Miscellaneous parasitic losses are taken to be approximately 2% of gross power production. They include transformer losses, transmission losses, and other in-plant electrical requirements not included anywhere else.

Heat Exchanger Calculations

Heat exchanger sizing and cost are determined by calculating the surface area required and estimating the cost by using the cost per square foot from Holt's heat exchanger cost database. The surface area is calculated using standard heat transfer methods, including the commercially available heat exchanger design and rating program HTC-STX. Exchanger costs are adjusted to compensate for high turbine inlet pressures (above 250 psia) using a published method (Purohit, 1983).

APPENDIX B

SYNCHRONOUS TURBINE-GENERATORS

Introduction

This section of the report was prepared by Ken Nichols of Barber-Nichols.

The overall goal of this portion of the NGGPP study is to investigate turbine-generator subsystems that are reliable, efficient, and low cost. To achieve reliable operation, turbines that function at generator (synchronous) speeds with small rotor diameters will be studied. This keeps the turbine operating stresses to a minimum. Such a design eliminates the requirement for a speed reducing gearbox, thereby reducing cost and improving reliability and efficiency.

Efficient operation will be achieved by selecting the near optimum number of parallel units and the near optimum number of turbine stages. By careful selection of these items, each stage will be operating very near the conditions required for peak efficiency.

The goal of low cost will be achieved as follows: the gearbox is eliminated, the turbine design allows the use of low strength materials due to the low operating stress, and a common design can be used for various sites through minor modifications and by adding or deleting stages.

Therefore, the goals are quite clear as is the means of achieving them. However, the path to be followed is to use five sites with two different working fluids (commercial isobutane and a 94-6 isobutane-heptane mixture) as the models for the turbine design. Assuming a trade-off between two different numbers of parallel units and an average of approximately four stages per unit, there are approximately 160 different turbine wheel designs that should be calculated and studied to come up with the common set of hardware which will have up to five or six stages. With the limited budget of this program, the final result will be a reasonable first cut towards the ultimate, optimized design.

Technical Background

Specific Speed and Specific Diameter

Given, for each site and working fluid, the working fluid flow rate and the head drop (enthalpy change) available to the turbine, the task at hand is to select the number of turbines in parallel, the number of turbine stages per turbine, and the diameter/blade length per stage to achieve good efficiency along with relatively small wheel diameters. The turbine shaft speed is to be the same as that of the generator (no gearbox). For the machine sizes (power output) under consideration, generator shaft speeds of 3,600 rpm are available (2-pole generators) and will be the only shaft speed considered for this study.

To bring together these various parameters and correlate them with turbine performance and size, it is expedient to utilize some similarity parameters. The particular parameters to be utilized are the specific speed and specific diameter. These, in turn, are correlated with turbine performance (efficiency) through widely available charts. Just how these parameters are used is described below.

Through the technique known as dimensional analysis, the similarity parameters specific speed, Ns, specific diameter, Ds, Reynolds number, Re, and Mach number, Ma, are derived and serve as convenient parameters for presenting the performance of turbomachines. These four parameters are sufficient to completely describe the performance of geometrically similar turbomachines. For a given volume flow rate and for a given head change through a turbomachine, specific speed is a number indicative of the rotational speed and specific diameter is a number indicative of the rotor diameter or the size of the machine. Reynolds number expresses the ratio of inertia force to viscous force and reflects the properties of the fluid flowing through the machine. Since turbines normally operate with compressible fluids, Mach number is used as the fourth similarity parameter.

It is difficult to present the performance of any turbine as a function of four parameters at one time. Fortunately, two of these variables, namely Reynolds number and Mach number, have only a secondary effect on turbine performance. More significantly, if the Reynolds number is above 10^6 for turbines, the effect of Reynolds number is very nearly constant which eliminates it as a variable. Likewise, if the Mach number of the turbine is less than or near 0.5, the compressibility effects are negligible which eliminates it as a variable. Turbine performance can then be presented as a function of the two variables, specific speed, Ns, and specific diameter, Ds. Therefore, this is the approach taken to analyze the performance of the multi-stage turbines for commercial isobutane and for the 94-6 mixture.

To better understand the basis of presenting turbine performance as a function of similarity parameters, an example and discussion of a performance plot for turbines will be covered below. The correlating similarity parameters, as discussed previously, are the specific speed Ns and specific diameter Ds

where $$Ns = \frac{NQ^{1/2}}{H^{3/4}}$$

and $$Ds = \frac{DH^{1/4}}{Q^{1/2}}$$

N	=	rotational speed, rpm
Q	=	exit volume flow rate, ft^3/sec
Had	=	adiabatic head, feet
D	=	rotor diameter, feet

It will be noted that the specific speed and specific diameter are not dimensionless in the form presented above; however, the parameters can be truly dimensionless when reduced to a form using angular velocity. An example of a typical specific speed, specific diameter performance correlation for full admission axial turbine is shown in Figure B-1. A good approximation of the turbine efficiency may be obtained by using an applicable curve such as Figure B-1. As can be seen from Figure B-1, turbine efficiencies greater than 80% are possible at specific speeds between 40 and about 250.

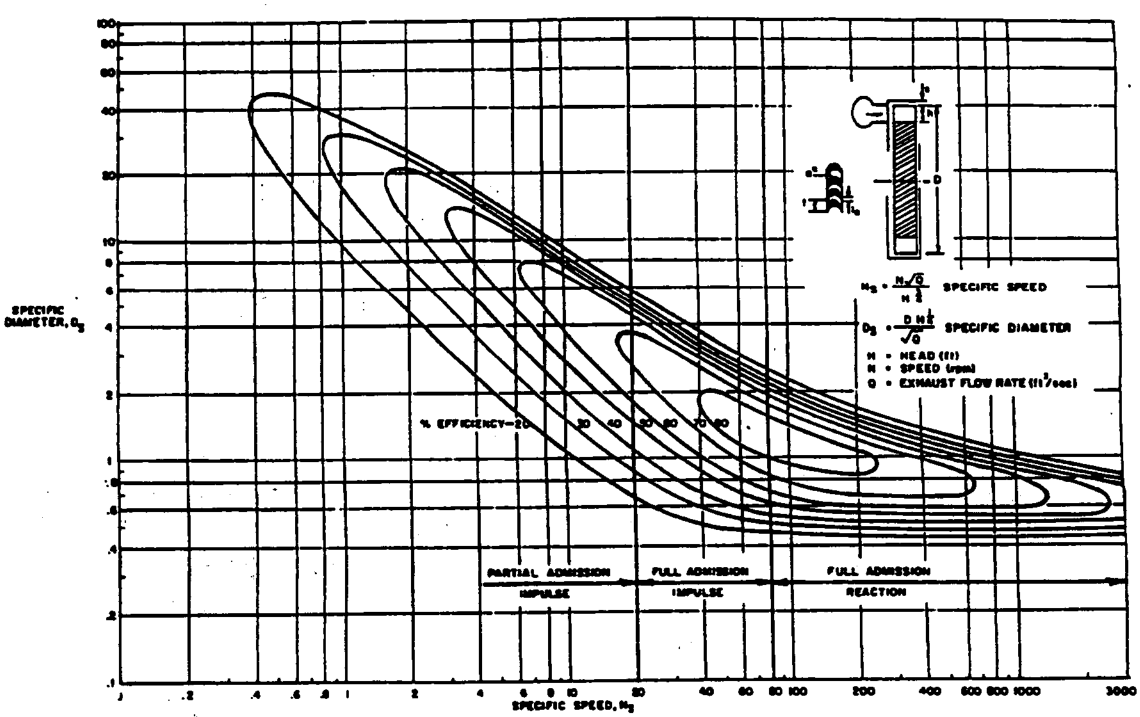

Figure B-1
Optimized Performance Chart, Axial-Flow Turbines

Ns-Ds diagrams are used to determine the performance of a turbine for a specific application as follows. The starting point is to calculate the isentropic head drop across the turbine. For the present study employing multiple stage units, this requires the selection of the number of stages and the selection of how the total head available is distributed over the various stages. The volume flow is then determined. For turbines, the volume flow used in the Ns and Ds calculation is taken at the exit of the subject turbine stage. A turbine efficiency is first assumed and then the exhaust specific volume can be determined. This, along with the mass flow, then gives the total flow volume. For the present study, the volume flow can also be adjusted by selecting different numbers of units in parallel. The volume flow for each stage, of course, is also a function of how the head is split up over the various stages.

Now, since it is desired to achieve a certain minimum efficiency, one can determine on the Ns-Ds diagram the lowest value of Ns that provides this efficiency. For the present application, the rotational speed is known to be 3600 rpm. Therefore, once an exit volume flow rate has been calculated (using an assumed efficiency) and the head drop determined, the specific speed can be calculated. The turbine efficiency is then found from the Ns-Ds diagram. If it is substantially different from the assumed efficiency, the new efficiency is used to recalculate the exit volume flow rate and the process above repeated. The Ds value from the Ns-Ds diagram will determine the turbine rotor diameter. This procedure is much simpler than evaluating the entire detailed equations that govern turbine performance and much time can be saved during a preliminary design phase when selecting or matching turbine components.

More detailed Ns-Ds diagrams also give blade height-to-diameter ratios as a function of specific speed and specific diameter. Thus, the blade height and blade root diameter can be determined. This information can be used to select turbine rotor sizes which best fit a number of turbine applications with the minimum of hardware components and variations.

Thermodynamic Data Base

At the beginning of this portion of the study, it was discovered that the fluid state point programs used by Barber-Nichols and that used by CE Holt Co. (Holt) produced slightly different results. The basis for Holt data is the Starling correlation while that used by Barber-Nichols is the National Institute of Standards and Technology (NIST 14).

While the saturation pressure-temperature values vary somewhat between the two data bases, the enthalpy drop for the turbine varies no more than just over 4%. In most cases, the variation is 2% or less. This certainly is within the accuracy of the study.

As a means of establishing a comparison of the two data bases, the enthalpy difference was determined for similar situations. The data generated by Holt was used as the basis. Thus, for the sites where subcritical cycles were used, the saturation temperature was determined for the pressure as stipulated in Holt data. This temperature was then rounded up to the next integer value to provide a slight amount of superheat. This pressure and temperature was then used as the turbine inlet condition.

An isentropic expansion to the turbine exit pressure as specified in the Holt data was used to establish the isentropic exit condition. Then an 85% expansion efficiency was used (the same as used by Holt to determine the actual turbine exit enthalpy. The enthalpy difference could then be compared with the data from Holt. The actual turbine exit temperature could also be computed and compared.

For the supercritical cases, the pressure and temperature, as specified by Holt data, was used as the turbine inlet condition. The remainder of the computation was identical to the computations described above.

This data and the results are shown in Table B-1. The enthalpy differences are greater for the commercial isobutane than for the 94-6 mixture. For commercial isobutane, the differences are 2% to just over 4%. For the mixture, on the other hand, the difference for three of the four cases is much less than 1%. The other case is about 1.5% different.

These differences are well within the accuracy and assumptions of this study. Therefore, all the Barber-Nichols turbine analysis was completed using the National Institute for Standards and Technology data base. This particular form of the data was a software package named NIST 14.

Baseline Case - Vale

The Vale resource has a temperature that is about in the middle of the range being considered in this study. Therefore, these conditions were selected for use with a binary plant using commercial grade isobutane to provide a baseline comparison between radial inflow and axial flow turbine designs.

Radial Inflow Turbine

The radial inflow turbine has performance equivalent to that of an axial flow turbine over a specific speed range of 45 to 110. Above a specific speed of 95, performance drops off rapidly. For the large multi stage units that are being considered for this project, radial inflow turbines have disadvantages that, in addition to lower maximum allowable specific speed, include very large, expensive housings that are required to accommodate the interstage ducting.

Table B-1
Data Base Differences

CASE	TDEW DEG F	T IN DEG F	P IN PSIA	H IN BTU/LB	S IN BTU/ LB-F	T-PRIME DEG F	P OUT PSIA	H PRIME BTU/LB	BN DEL H BTU/LB	BH DEL H BTU/LB	DIFF %	T OUT DEG F	BN DEL T DEG F	BH DEL T DEG F
IC4														
THS	191.8	192.0	235	-966.725	1.18902	101.50	57.2	-990.894	20.544	20.954	-1.96	109.6	82.4	78.7
RR	222.4	223.0	325	-960.394	1.19110	107.25	60.6	-988.717	24.075	24.589	-2.09	116.7	106.3	101.3
V	SC	293.5	600	-954.507	1.18864	109.70	67.0	-988.360	28.772	30.052	-4.26	120.8	172.7	167.0
SV	SC	353.0	850	-920.868	1.22632	157.00	66.0	-966.290	38.609	39.597	-2.50	171.3	181.7	179.1
GM	SC	340.0	800	-929.539	1.21654	146.65	68.9	-971.443	35.618	36.674	-2.88	159.9	180.1	176.8
94-6														
THS	230.6	231.0	250	-935.799	1.20911	137.70	55.0	-963.049	23.163	23.125	0.16	146.3	84.7	81.7
RR	244.8	245.0	300	-931.879	1.21037	142.40	58.3	-961.237	24.954	24.589	1.48	151.6	93.4	92.5
V	282.6	283.0	475	-931.143	1.20211	139.80	64.3	-964.428	28.292	28.403	-0.39	146.8	136.2	134.5
SV	SC	323.0	700	-935.686	1.19063	129.62	63.4	-973.153	31.847	32.036	-0.59	136.1	186.9	181.9
(THS: Thermo Hot Springs, RR: Raft River, V: Vale, SV: Surprise Valley, GM: Glass Mountain)														
COL#	1	2	3	4	5	6	7	8	9	10	11	12	13	14

NOTES
SC IN COL 1 DESIGNATES SUPERCRITICAL
PRIME DESIGNATES PROPERTIES AFTER ISENTROPIC EXPANSION
BN IS BARBER-NICHOLS
BH IS BEN HOLT CO
DEL H IS FOR 85% TURBINE (COL 9 & 10)

The limitation on the specific speed for radial inflow turbines is very severe and makes it difficult to use multiple stage units with reasonable efficiency. This limitation is a result of the losses associated with this specific hardware configuration.

Baseline Radial Inflow Turbine

A baseline radial turbine design was investigated for the Vale resource for a binary plant with a nominal 62,000 kW gross turbine output (the hydrocarbon flow rate of 7,483,717 pounds per hour, as specified, was used in the computation). To eliminate the need for a gear box, a turbine shaft speed of 3600 rpm is specified. The maximum turbine rotor size is limited to a 36 inch diameter because of the limiting specific speed of about 95 to 110. For this very preliminary analysis, a number of simplifications have been made which include using equal pressure ratio across each stage and assuming negligible pressure drop between each stage.

To satisfy these design parameters, it is necessary to use six units operating in parallel with each unit having four radial flow turbine stages. To minimize the number of turbine housings and generators, a split flow design is used in which the flow enters the middle of the turbine housing and is divided so it flows through two sets of turbine stages. With this design, three housings and three generators will be required. This design has the added advantage of balancing thrust loads on the turbine shaft. Such a possible arrangement is shown in Figure B-2.

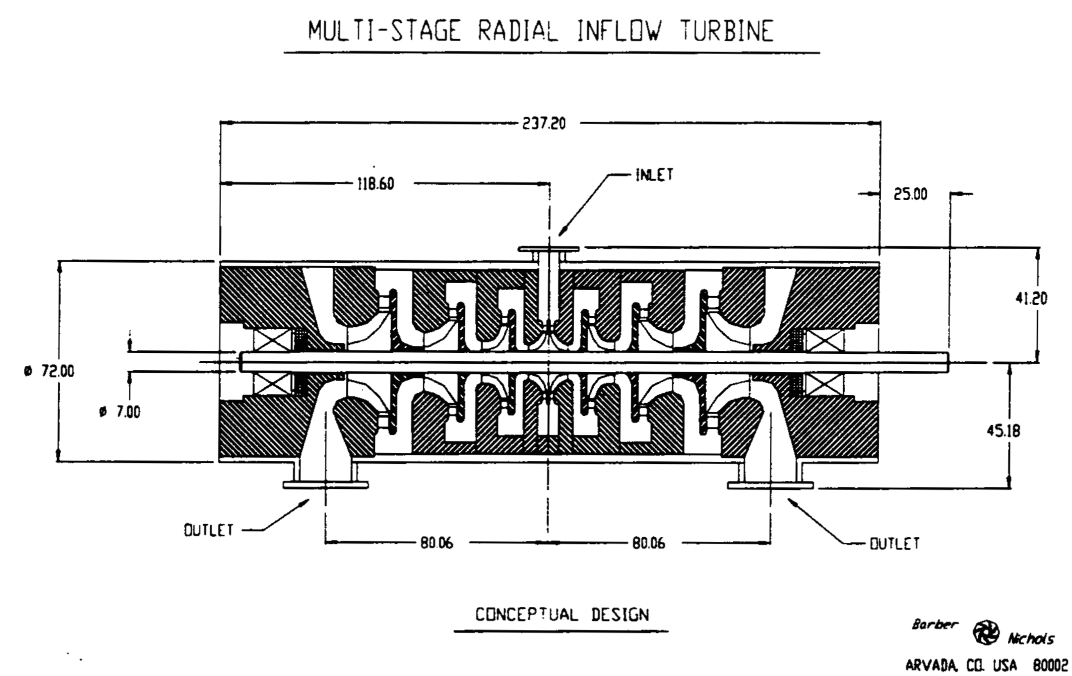

Figure B-2
Multi-Stage Radial Inflow Turbine

NASA Lewis Research Center did an experimental performance evaluation of a radial inflow turbine over a range of specific speeds[1]. This range extended from 72 to 108. They used a single turbine rotor with four different nozzles designed for 50, 75, 100 and 125 percent of design flow.

1 "Experimental Performance Evaluation of a Radial In-Flow Turbine Over a Range of Specific Speeds" by Milton G. Kofskey and Charles A. Wasserbaver, NASA TN D-3742, August 26, 1966.

The maximum efficiency (total-to-static) occurred with the 75% nozzle and was .87 at a specific speed of 86. This was four percentage points higher than the .83 obtained for the turbine with the 100% design flow nozzle at a specific speed of 95.6. At the design blade-jet velocity ratio, the lowest total-to-static efficiency of .77 was obtained at a specific speed of 108.

From these results, it appears that the peak efficiency range of radial inflow turbines occurs at specific speeds ranging from about 80 to 90. Figure B-3 shows the variation of turbine losses with specific speed at equivalent design speed and pressure ratio. From this figure it is clear that the specific speed of a radial inflow turbine should be limited to the range of about 80 to 90 for efficiencies above 85%.

Figure B-3 is for a specific turbine designed, built and tested by NASA. It illustrates the narrow specific speed range over which radial inflow turbine efficiency can be expected to exceed 80%.

Table B-2 shows some of the key design and performance parameters for a single set (one of six) of radial turbine stages. The efficiencies reported in this table are from more generalized Ns-Ds diagrams and, as a result, are somewhat higher than those reported in the NASA study.

There are three major disadvantages with a radial turbine for this application. First, the specific speed limitation requires the use of 6 parallel units to limit the maximum stage specific speed to 110. Second, the required shaft diameter may be larger than the eye diameter. Third, radial flow turbines require housings that are large and expensive compared to a comparable capacity axial unit.

With radial inflow turbines, it is especially important not to have the specific speed be too large as the turbine efficiency drop off with increasing specific speed is quite sharp. Therefore, additional units in parallel must be used to decrease the flow rate in each unit to keep the specific speed down. Note that in the above table the specific speed increases with each successive stage. The final stage is the one that must be limited in flow rate. An alternative would be to again split the flow, but this brings in added cost due to more rotors and larger housings.

The smallest eye diameter for the design listed above is 10 inches. For this particular turbine, the required shaft diameter is 7.5 inches from a torque carrying capacity standpoint. Using this shaft diameter and the 10-inch eye (OD) diameter, a blade height of 1.25 inches would be allowed. This is very small. Even more important, from a practical standpoint, the shaft should be sized much larger, probably in the 15-inch diameter range. This is due to critical speed calculations and also due to shaft deflection. If two bearings are used and all (eight) rotors are spaced between them (as shown in Figure B-2), the total bearing

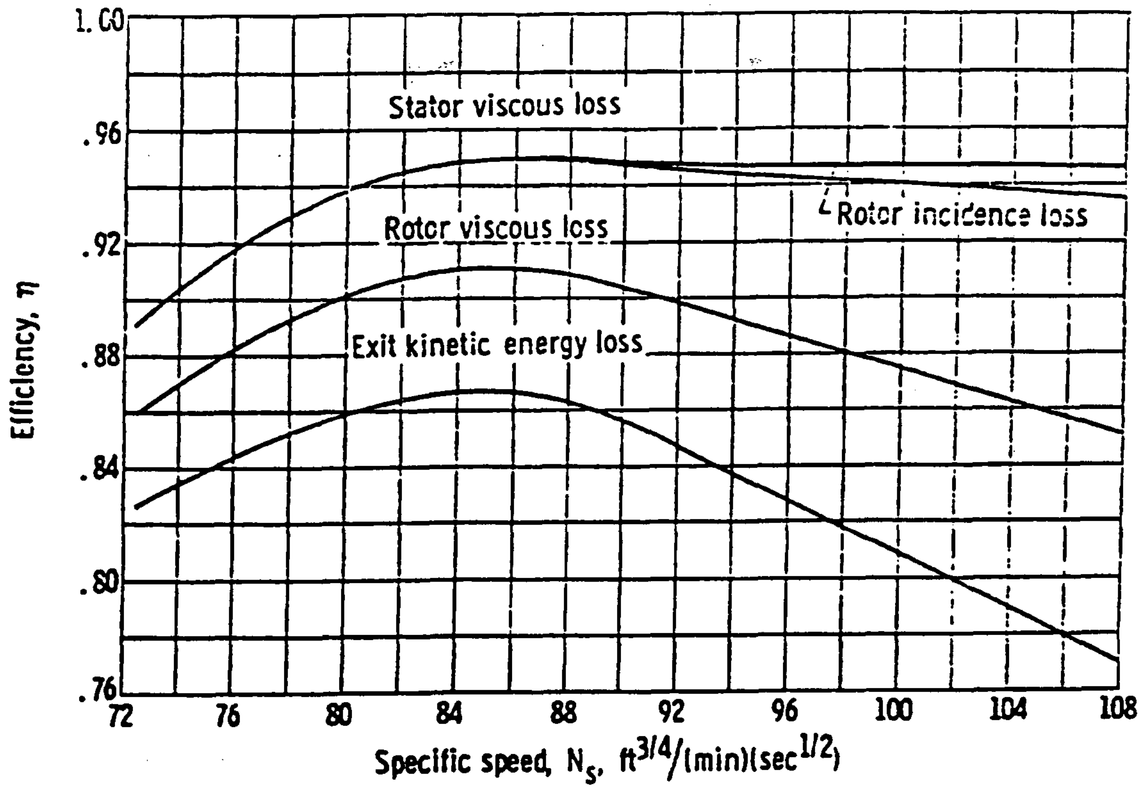

Figure B-3
Back to Back Axial Flow Turbines

span will be over 16 feet. To run reasonable clearances between the rotor and housing minimal shaft deflection can be allowed. Therefore, a larger shaft diameter is required for stiffness. An alternative would be to place more bearings on the shaft between the various rotor stages. This greatly complicates the design and adds significant costs.

The third disadvantage, a large, complicated housing results in high cost. The radial turbine housing must provide interstage passages to duct the flow from the eye of one stage to the nozzle plenum of the next stage. These interstage passages significantly increase the length of the radial turbine assembly compared to an axial turbine. The radial turbine also requires a large diameter nozzle plenum and nozzle ring for each stage. The outside diameter of the nozzle plenum for the fourth stage is approximately 72 inches.

Table B-2
Radial Flow Turbine Parameters for One of Six Parallel Units

	Stage 1	Stage 2	Stage 3	Stage 4
T(in) (F)	293	229	184	149
p(in) (psia)	600	347	200	116
Flow Rate (out) (ft^3/s)	76	155	286	502
Head (ft-lbf/lbm)	5170	6580	7320	7600
Ns	51	61	77	99
Rotor Tip Dia (in)	21	26	30	36
Eye Diameter	10	15	21	25
Nozzle Plenum OD (in)	42	52	60	72
Efficiency	.81	.82	.84	.85
Power (HP)	2650	3420	3866	4070

Axial Flow Turbine

An axial turbine design was also investigated for the Vale resource conditions. The major design parameters for the axial turbine are: shaft speed - 3600 rpm, maximum stage specific speed -300, and equal pressure ratio across each stage. The axial turbine has a much higher allowable specific speed (as discussed in Section 2.4). As a result of the larger allowable specific speed, only four parallel units are required for the axial turbine as opposed to six for the radial turbine. The axial turbine will use the same split flow housing design that was described above. Consequently, only two turbine housings and two generators will be required for this resource.

Table B-3 shows some of the key design and performance parameters for a single set (one of four) of axial turbine stages.

A possible arrangement of one set of these axial flow turbines (two turbines back to back) is shown in Figures B-4 and B-5. This is a much more compact and considerably less complex design than the radial inflow turbine shown in Figure B-2.

Table B-3
Axial Flow Turbine Parameters for One of Four Parallel Units

	Stage 1	Stage 2	Stage 3	Stage 4
T(in) (F)	293	229	184	149
p(in) (psia)	600	347	200	116
Flow Rate (out) (ft^3/s)	114	232	428	752
Head (ft-lbf/lbm)	5170	6580	7320	7590
Ns	63	75	94	121
Rotor Tip Dia (in)	23	28	34	40
Efficiency	.83	.84	.85	.86
Power (HP)	4040	5203	5870	6170

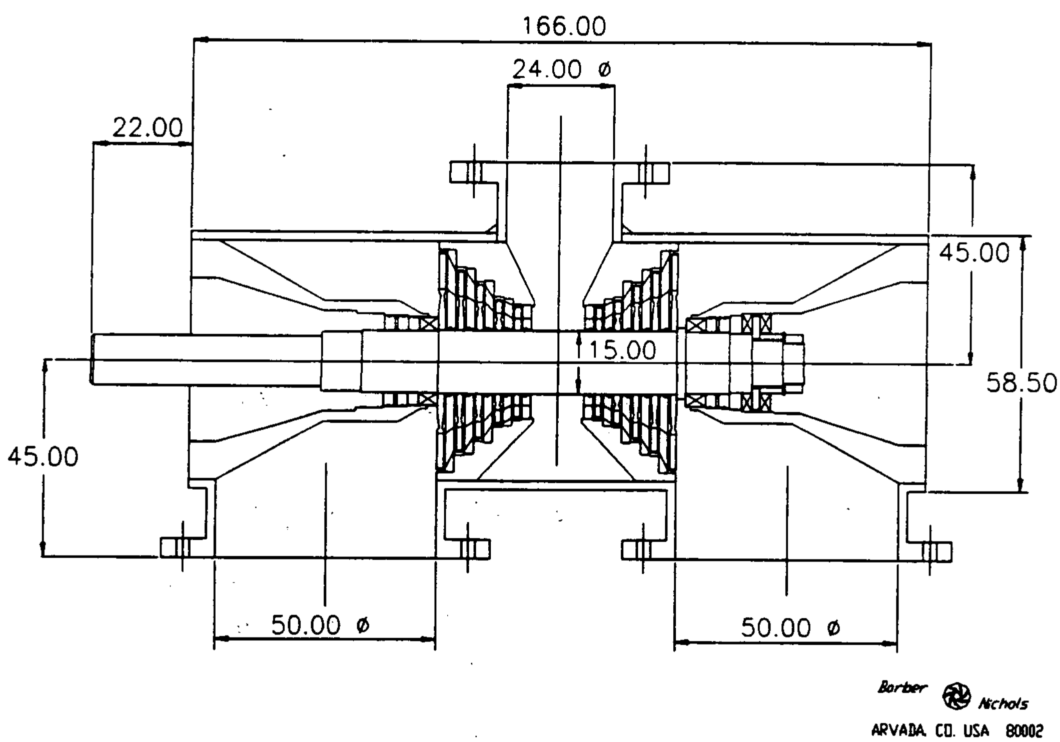

Figure B-4
Axial Flow Turbine: Typical Dimensions

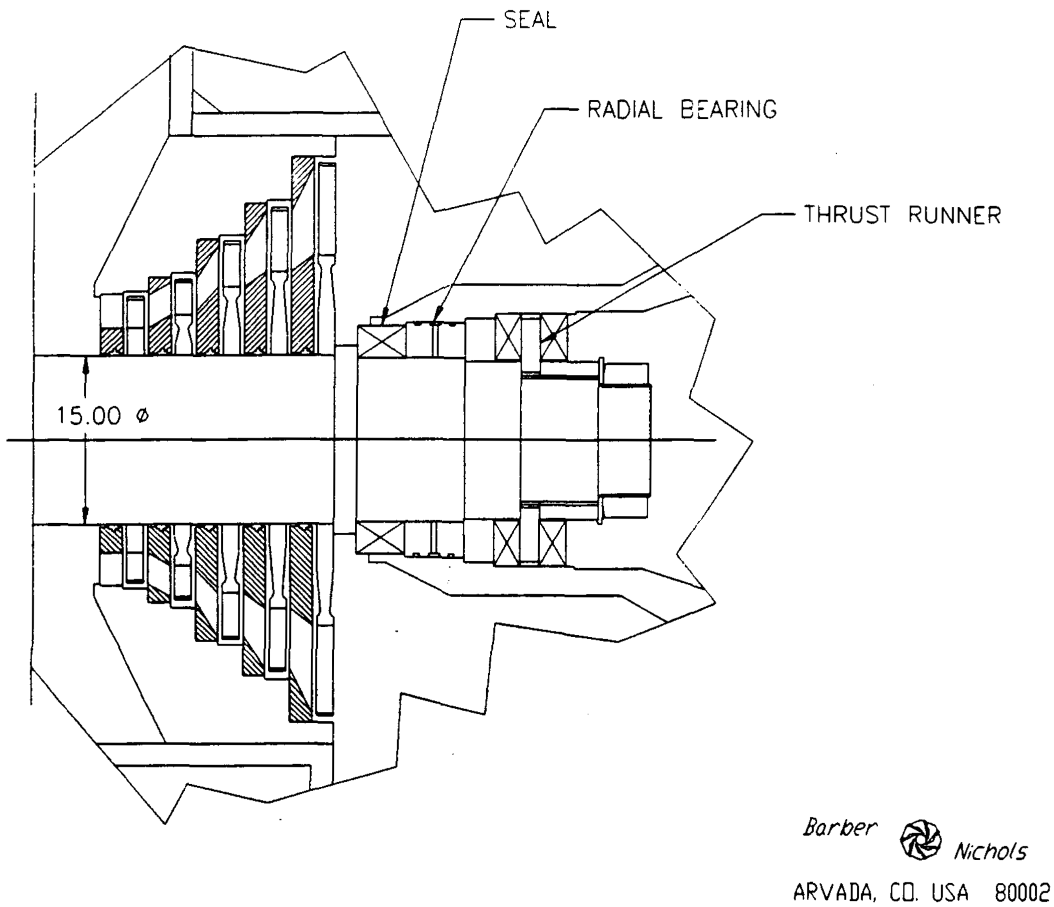

Figure B-5
Axial Flow Turbine Detail

The axial turbine has a minor performance advantage with stage efficiencies being slightly higher than the radial turbine. The major advantage of the axial turbine is cost. The axial design only requires two turbine-generator units where the radial design requires three. Furthermore, the housing for the radial design is much more complex and has a larger diameter due to plenum arrangement for radial inflow turbines. Furthermore, there is a real conflict between shaft size and the eye diameter for the radial inflow turbine that can only be overcome through the use of a very costly design, namely, multiple bearings. On the basis of these observations, the axial turbine design was selected as the baseline for the other resource conditions studied. Detailed cost information for the axial turbine will be provided in the following sections.

Axial Flow Conceptual Design

To define an economical multiple stage axial flow turbine for the various sites and working fluids, the goal was set to devise a common design that could be used in all cases. This was accomplished by defining a turbine that could have stages removed or added as necessary depending on the particular site conditions. Additionally, for the high resource temperature sites which have a smaller hydrocarbon flow rate, fewer machines can be used in parallel.

To identify this generic fits-all turbine, it was first necessary to look at the near optimum turbine configuration for each site. This was done for the isentropic head as described in Section 3.0 for each site along with the hydrocarbon flow rates provided by Holt. Various numbers of units in parallel were assumed for each site along with differing numbers of stages per unit.

Once this multitude of data was compiled, it was possible to sort through and discover where common rotor sizes were required. Some minor adjustments were made and a final design configuration was chosen. This final design is five stage turbine with the following rotor configuration:

Stage	Rotor OD, in	Blade Ht
1	22	2.2
2	28	2.8
3	36	3.6
4	41	5.4
5	48	7.8

With this design, it is possible to reasonably fit the requirements for each site. This is shown in Table B-4. All of these turbines operate at 3,600 rpm as generators were available in that speed for the power outputs being considered.

Performance

Once the turbine configuration was known, it was possible to determine the stage efficiencies and then compute the power output. This was done and the result is shown in Table B-5. Note that the power output reported here is not necessarily the optimum value as a number of assumptions have been made. One of the more significant assumptions was to use equal pressure ratios for each stage. That was expedient, but could be fine tuned to produce more power.

In the thermodynamic data section, a discussion of the impact of different thermodynamic data bases was provided. As a means of comparing the present results for the axial flow turbine (Table B-5) to Holt baseline, single stage, radial inflow turbine, Table B-6 is provided. This table uses the same format as Table B-5 for easy comparison. It will be noted that the axial flow computations

Table B-4
Turbine Staging

Site/Fluid	No. Parallel Turbines	Number Generators	Turb. Stages Used
Thermo Hot Springs, IC4	4	2	3-4-5
Raft River, IC4	4	2	2-3-5
Vale, IC4	4	2	1-2-3-4
Surprise Valley, IC4	2	1	1-2-3-4-5
Glass Mountain, IC4	2	1	1-2-3-4-5
Thermo Hot Springs, 94-6	4	2	3-4-5
Raft River, 94-6	4	2	3-4-5
Vale, 94-6	4	2	2-2-3-4
Surprise Valley, 94-6	4	2	2-3-4-5

generally provide higher power output (for the same hydrocarbon flow rate) even though the enthalpy differences for the Barber-Nichols data base were almost always lower. However, the axial flow computations have the benefit of actual turbine efficiency calculations rather than the assumption of an efficiency of 85%, as used in the Holt computation, and also a better generator efficiency. The gearbox loss for the radial inflow turbine is a real difference as the multi stage axial flow machine does not need the speed reducer.

The turbine efficiency difference could be real as the axial flow turbine can operate at a higher specific speed. This was discussed in detail in the discussion of the baseline case, Vale, Oregon.

Cost

Given the design discussed above, it was possible to price the various components. Because of the common design, most of the items were used at all sites. The significant difference between the various sites was the use of a single, large generator with two turbines for Surprise Valley and Glass Mountain with isobutane. All other sites used two generators and four turbines. Therefore, the two single generator sites are significantly less expensive on a cost per unit power basis.

Table B-5
BN Power Output

Site/Fluid	Hyd Flow Rate, lb/hr	Isentrop. Head, Btu/lb	Isentrop. Pwr, kW	Mech Loss, kW	Turbine Effic.	Turbine Power, kW	Gearbox Eff.	Generator Eff.	Net Output Pwr, kW
THS, IC4	10,536,220	24.169	74,612	100	86.9%	64,738	100%	97.7%	63,249
RR, IC4	8,795,528	28.323	72,990	100	86.0%	62,672	100%	97.7%	61,230
V, IC4	7,483,717	33.853	74,230	100	85.5%	63,366	100%	97.7%	61,909
SV, IC4	5,708,697	45.422	75,974	50	86.2%	65,440	100%	98.0%	64,131
GM, IC4	6,168,632	41.904	75,737	50	86.5%	65,462	100%	98.0%	64,153
THS, 94-6	9,366,942	27.25	74,787	100	87.0%	64,965	100%	97.7%	63,471
RR, 94-6	8,645,803	29.358	74,370	100	86.0%	63,858	100%	97.7%	62,389
V, 94-6	7,726,822	33.285	75,355	100	86.0%	64,705	100%	97.7%	63,217
SV, 94-6	7,113,635	37.467	78,092	100	86.9%	67,762	100%	97.7%	66,203

THS: Thermo Hot Springs, RR: Raft River, V: Vale, SV: Surprise Valley, GM: Glass Mountain

Table B-6
BH Power Output

Site/Fluid	Hyd Flow Rate, lb/hr	Isentrop. Head, Btu/lb	Isentrop. Pwr, kW	Mech Loss, kW	Turbine Effic.	Turbine Power, kW	Gearbox Eff.	Generator Eff.	Net Output Pwr, kW
THS, IC4	10,536,220	24.652	76,102	0	85.0%	64,687	98%	95.92%	60,806
RR, IC4	8,795,528	28.928	74,550	0	85.0%	63,367	98%	95.92%	59,565
V, IC4	7,483,717	35.349	77,511	0	85.0%	65,884	98%	95.92%	61,931
SV, IC4	5,708,697	46.585	77,919	0	85.0%	66,231	98%	95.92%	62,257
GM, IC4	6,168,632	43.146	77,982	0	85.0%	66,284	98%	95.92%	62,307
THS, 94-6	9,366,942	27.206	74,666	0	85.0%	63,466	98%	95.92%	59,658
RR, 4-6	8,645,803	29.224	74,029	0	85.0%	62,925	98%	95.92%	59,149
V, 94-6	7,726,822	33.415	75,650	0	85.0%	64,303	98%	95.92%	60,444
SV, 94-6	7,113,635	37.689	78,555	0	85.0%	66,772	98%	95.92%	62,766

THS: Thermo Hot Springs, RR: Raft River, V: Vale, SV: Surprise Valley, GM: Glass Mountain

In comparing the various sites where two generators were used, there are some minor differences due to the different turbine stages used. These minor differences, along with the minor power output variation, produce nearly level costs per unit power. For the two generator systems, the costs are about $165 per kilowatt. The single generator systems are slightly above $100 per kilowatt.

The baseline power unit can be priced, less any stages, for the single generator (65 MW) and double generator (32 MW) cases. Pricing it without stages then allows the price addition of the stages required for each site to establish the site dependent price. Note that these costs were generated assuming this was a fully developed turbine. There are no costs included for design engineering or developments costs.

The turbine without stages have the same hardware for both the single or double generator systems and therefore have the same cost. The generators, of course, have difference costs. For the two sizes chosen, the smaller machine costs $44/kW while the larger machine costs $28/kW. This fairly large difference is due to the particular frame sizes that these machines happen to fall into. Other than the generator cost differences, there are only minor differences in the large and small system costs. The cost breakdown for these two sizes of systems are shown in Table B-7.

The cost of the various stages is dependent on the number of blades (both rotor and stator) and the diameter of the stage. The costs for the stages of the particular machines for each site considered here is shown in Table B-8.

The costs in Table B-7 for the baseline unit without stages can be combined with the stage costs given in Table B-8 to obtain the total cost. This is shown in Table B-9 along with the power output and the cost per unit power output.

Technical Risks

Designing and fabricating a multiple stage, axial flow turbine as described in this report is well within existing technology. There are no new material problems to overcome nor any extension of knowledge required.

The turbine described here is very similar to present day steam turbines with the exception of somewhat higher pressures than are normally encountered in geothermal service.

The real risk is one of economics. It would require a significant investment to design and construct such a device and to go through the normal development cycle. The company that undertook such an effort would need some form of assurance that there would be a reasonable sized market for such a machine or would need some form of financial assistance from industry/government to undertake this effort.

Table B-7
Power Unit Cost

Line	Cost Element	32 MW Note 1	65 MW Note 1
1	Turbine without stages	$1,435	$1,435
2	Turbine lube system	150	150
3	Generator	1,400	1,800
4	Generator lube system	85	100
5	Couplings	40	60
6	Turbine valve	350	350
7	Marketing, warranty, profit	1,000	1,500
8	Misc.	300	500
	Total	$4,760	$5,895

Table B-8
Cost of Stages

Stage	Pitch dia. inches	Number of blades	Cost Note 1
1	23	90	$111,760
2	25	95	$119,544
3	31	100	$132,640
4	35	105	$143,240
5	41	110	$154,480
Total			$661,664

Note 1: Cost includes rotor and stator

Table B-9
Power Unit Cost

Site/Fluid	Power out per unit, kW	No. of units required	Stages used	Stage cost	Turb/gen cost	Unit cost	Total Cost	Cost per unit of pwr
THS, IC4	31,624	2	3-4-5	430,360	4,760,000	5,190,360	10,380,720	164
RR, IC4	30,615	2	2-3-5	406,664	4,760,000	5,166,664	10,333,328	169
V, IC4	30,954	2	1-2-3-4	507,184	4,760,000	5,267,184	10,534,368	170
SV, IC4	64,131	1	1-2-3-4-5	661,664	5,895,000	6,556,664	6,556,664	102
GM, IC4	64,153	1	1-2-3-4-5	661,664	5,895,000	6,556,664	6,556,664	102
THS, 94-6	31,735	2	3-4-5	430,360	4,760,000	5,190,360	10,380,720	164
RR, 4-6	31,195	2	3-4-5	430,360	4,760,000	5,190,360	10,380,720	166
V, 94-6	31,609	2	2-2-3-4	514,968	4,760,000	5,274,968	10,549,936	167
SV, 94-6	33,102	2	2-3-4-5	549,904	4,760,000	5,309,904	10,619,808	160

THS: Thermo Hot Springs, RR: Raft River, V: Vale, SV: Surprise Valley, GM: Glass Mountain